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A Heat Pump Waste Heat Recovery System for the Dry Cleaning and Laundering Service Sector

James F. Hickling Management Consultants Ltd.

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A HEAT PUMP WASTE HEAT RECOVERY SYSTEM FOR THE DRY CLEANING AND LAUNDERING SERVICE SECTOR

for the

Heat Pump Program
Division of Energy Research
National Research Council Canada

prepared by
James F. Hickling Management Consultants Ltd.
Ottawa, Ontario

September, 1984

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EXECUTIVE SUMMARY

This report details the installation and service demonstration of a heat pump waste heat recovery system in a small laundering and dry cleaning plant.

The successful outcome of this heat pump application is largely determined by site specific criteria and the selection of a suitable heat pump. The cost of the system and installation must be kept to a minimum in order to make the payback period as short as possible and therefore attractive to the small businessman.

The heat pump waste heat recovery system was installed at Betty Brite Cleaners, 218 Laurier Avenue East in Ottawa. The system consists of a heat pump with a waste water storage tank and a fresh water storage tank each of 545 litre capacity. The water is circulated through the closed loop system until a desired temperature has been reached in the fresh water storage tank. The waste heat recovery system has been tested under operating conditions at Betty Brite Cleaners and is currently providing all of the hot water demands of the plant at a coefficient of performance (COP) of approximately 2.6. The actual operating system was compared with the predicted system modelled on a TRNSYS computer simlulation and performed the same way as predicted. Break Even Capital Cost (BECC) calculations were performed on the actual operating system. The resulting payback period of 3.5 years should ensure market penetration of the system and further reduction in the capital cost.

A complete analysis of the dry cleaning and laundering service sector has shown that the potential penetration and energy savings which could be gained by the heat pump waste heat recovery system are very good. It is expected that a potential 17% energy savings could be accomplished for the total service sector and that a 0.1% total reduction in energy demand for the industrial sector in Canada is possible.

RÉSUMÉ À L'INTENTION DE LA DIRECTION

Le présent rapport décrit en détail la mise en place et le fonctionnement dans des conditions réelles d'un système de récupération de la chaleur perdue par pompe à chaleur dans une petite installation de blanchissage et de nettoyage à sec.

La réussite de cette application de la pompe à chaleur est déterminée en grande partie par des critères propres au site et par la sélection d'une pompe à chaleur appropriée. Le coût du système et de son installation doit être limité le plus possible afin que le délai de récupération soit aussi court que possible et que, par conséquent, le système soit une option attrayante pour les dirigeants de petites ou moyennes entreprises.

Le système de récupération de la chaleur perdue par pompe à chaleur a été installé chez Betty Brite Cleaners au 218 de l'avenue Laurier est, à Ottawa. Il consiste en une pompe à chaleur équipée d'un réservoir de stockage des eaux résiduaires et d'un réservoir de stockage de l'eau fraîche, d'une capacité de 545 litres chacun. L'eau circule dans un système à boucle fermée jusqu'à ce que la température voulue soit atteinte dans le réservoir de stockage de l'eau fraîche. Le système de récupération de la chaleur perdue a été mis à l'essai dans des conditions réelles de fonctionnement chez Betty Brite Cleaners et il satisfait actuellement à tous les besoins en eau chaude de l'installation de blanchissage et de nettoyage à sec et ce, avec un coefficient de performance (COP) d'environ 2,6. Le véribable système mis en place a été comparé à un modèle de système utilisé aux fins de la simulation sur ordinateur TRNSYS et sa performance s'est avérée conforme aux prévisions. Les calculs du coût en capital critique ont été effectués à l'aide du véritable système en place. Le délai de récupération obtenu, lequel s'établit à 3,5 ans, devrait assurer l'entrée sur le marché de ce système et réduire davantage le coût en capital.

Une analyse détaillée du secteur des services de blanchissage et de nettoyage à sec a démontré que les possibilités de pénétration du marché par le système de récupération de la chaleur perdue par pompe à chaleur et les économies d'énergie qu'il permettrait de réaliser sont très bonnes. On s'attend à ce que des économies d'énergie éventuelles de l'ordre de 17 p. 100 soient réalisées pour l'ensemble du secteur offrant ce genre de service et qu'une réduction totale de 0,1 p. 100 de la demande d'énergie de tout le secteur industriel au Canada soit possible.

1.0 INTRODUCTION

1.1 BACKGROUND

1.1.1 Program Development

The National Research Council, Division of Energy, has been seeking methods for the implementation of heat pumps in commercial applications. The objective of the program is to support the implementation of the National Energy Program by reducing fossil fuel consumption through substitution of these fuels for centrally generated electrical energy.

Under DSS 01SX.315S-1-2687-X, dated April 1, 1982, proposals were sought for novel applications of heat pumps in industrial processes and in particular those cases which are directed to specific plant applications. The general objectives of this project include:

- o the expansion of the use of heat pumps into areas without previous operating experience;
- o the identification of areas where equipment development is required;
- o the determination of design parameters which are important to waste heat recovery systems.

1.1.2 Commercial Dry Cleaning and Laundering Operations

There are approximatley 4,000 dry cleaning and laundering plants throughout Canada (800 in Ontario alone). About 60% of these small plants provide shirt laundering services. In addition to these plants, it has been estimated that there are an additional 400 power launderers operating in Canada.

Reliable statistics on this service sector were last published by Statistics Canada in 1976 and were based on a 1974 survey of this service sector. STATSCAN has not surveyed this sector since 1974 thus the market potential for energy saving devices can only be based on plants registered with their respective provincial associations.

Although capital costs for laundering equipment are high, dry cleaning establishments appear to be best suited to provide this service. Dry cleaning and laundering can be integrated such that steam required for pressing equipment and hot water for laundering can be provided by single high capacity boilers, generally oil or natural gas fired. James F. Hickling Management Consultants Ltd. (JFH) has inspected four such plants in the Ottawa area. JFH found that hot water is provided generally in the same manner:

- o makeup water from a boiler is used to generate steam at 275 kPa for direct use in pressing equipment;

- o steam is passed through a splitter valve with cold tap water to generate hot water at approximately 70°C. The hot water is used in industrial capacity washing machines. Waste water that contains dissolved soaps, soil, lint, etc. is dumped periodically during a wash cycle as waste in quantities which appear to be most suitable to waste heat recovery applications.

1.1.3 Waste Heat Recovery

There are two approaches that offer recovery of waste heat at moderate capital cost. The first involves the use of commercially available water-to-water heat exchangers for laundromat applications.

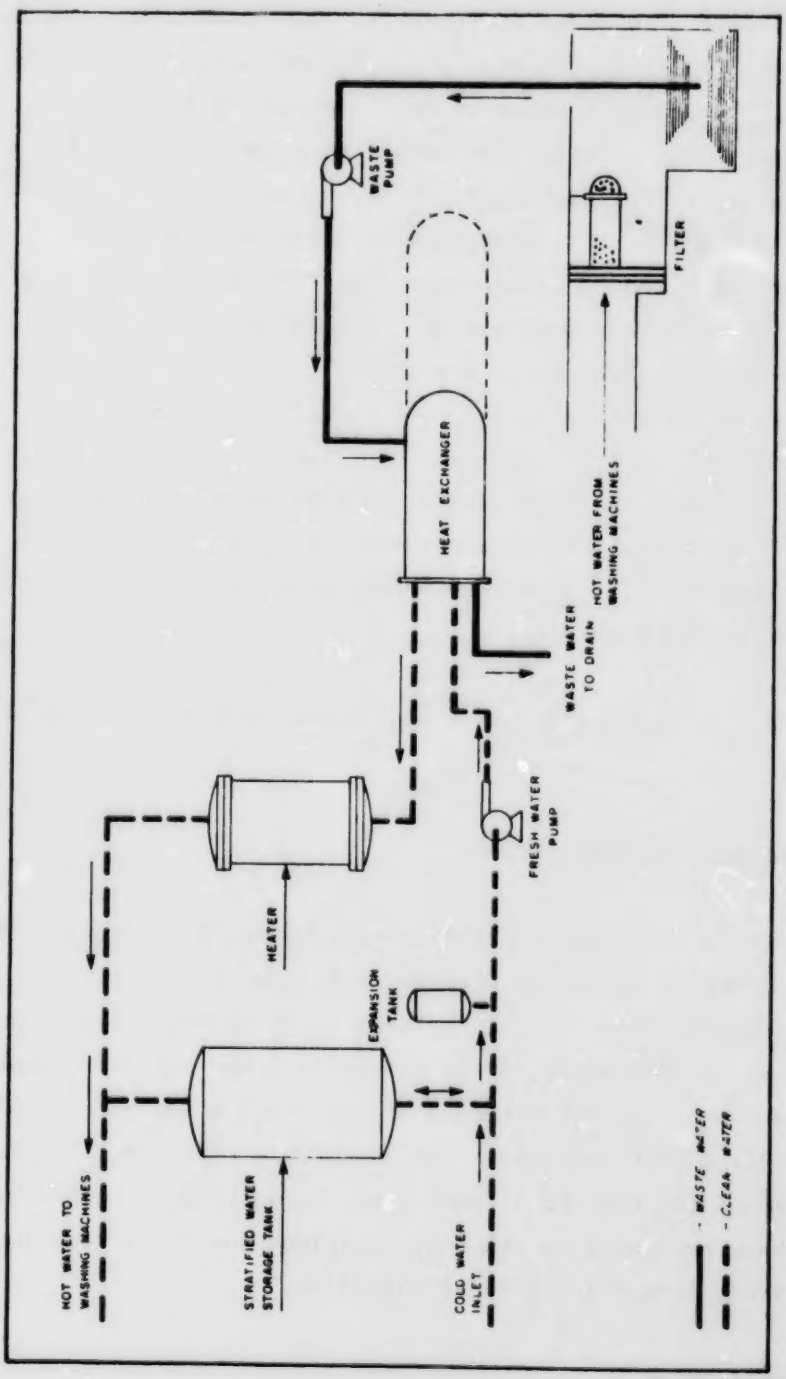


Fig. - 1 - TYPICAL WASTE HEAT RECLAMATION SYSTEM USING COUNTERFLOW HEAT EXCHANGER

A typical installation is shown in Figure 1. Hot water wash is dumped first to a channel leading to a sump pit. Before it reaches the pit it passes through a series of screens to remove excess lint and other solids. The dirty hot water is then pumped from the sump into the shell of the water-to-water heat exchanger. Fresh water is pumped through the heat exchanger and exits at temperatures normally at 38°C. This preheated fresh water can be either stored for use upon demand from the splitter valve with conventional boiler installation or it can be passed through a water heater where it is brought up to 82°C and then stored in a storage tank for direct use in the washing machines. These systems have been mainly used in power laundery applications. Small operations do not necessarily have the floor space and sump pits to accommodate such a system. Although heat exchangers are very large with these systems, some manufacturers do offer copper coil heat exchangers for installation in open sump pits (see Figure 2).

A second approach to economical waste heat recovery would involve the use of heat pumps.

1.2 HEAT PUMP WASTE HEAT RECOVERY SYSTEM

This report details the novel application of the use of heat pumps for waste heat recovery in small dry cleaning and laundering plants in Canada. Based on the relatively large quantities of waste heat and the cyclic nature of the operation, it is considered that the dry cleaning and laundering service sector offers a viable application for the use of heat pumps in waste heat recovery. The concepts of Figures 3 and 4 show typical approaches for the use of heat pumps in waste heat recovery. Such systems have been used in other applications for the production of domestic hot water from heat sinks as varied as:

- well water
- solar assisted sources
- salt ponds.

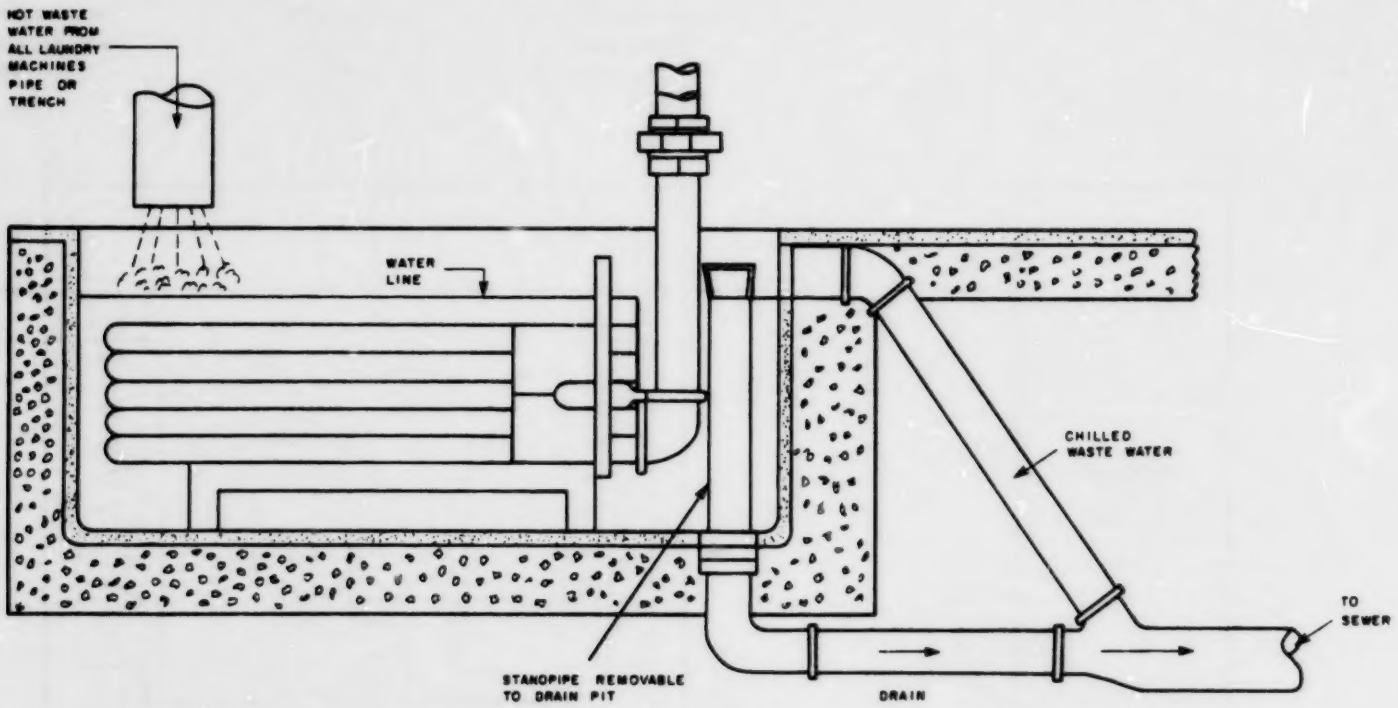


Fig. - 2 - PIT INSTALLED HEAT EXCHANGER

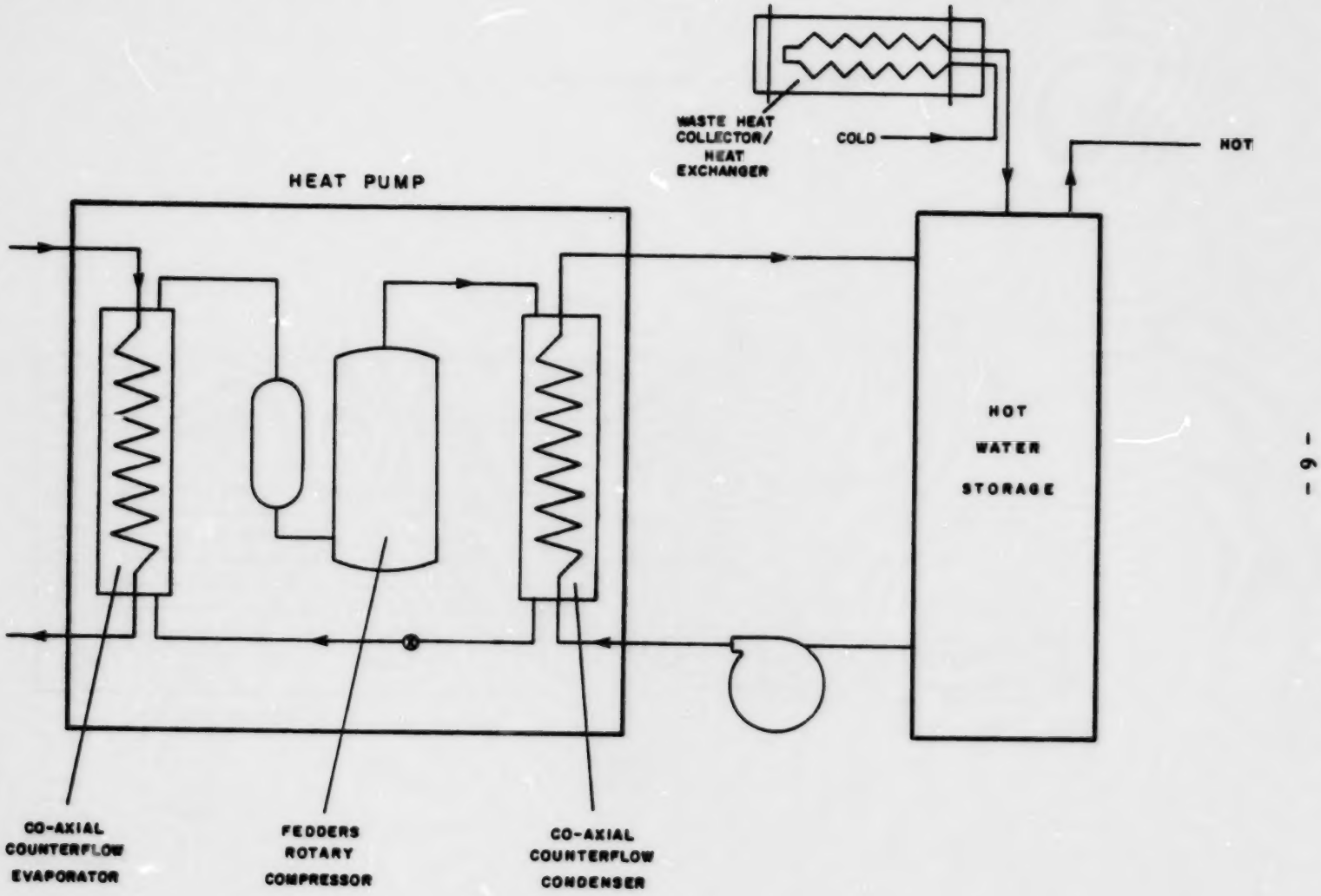


Fig.-3 - WASTE HEAT RECOVERY WITH HEAT PUMP

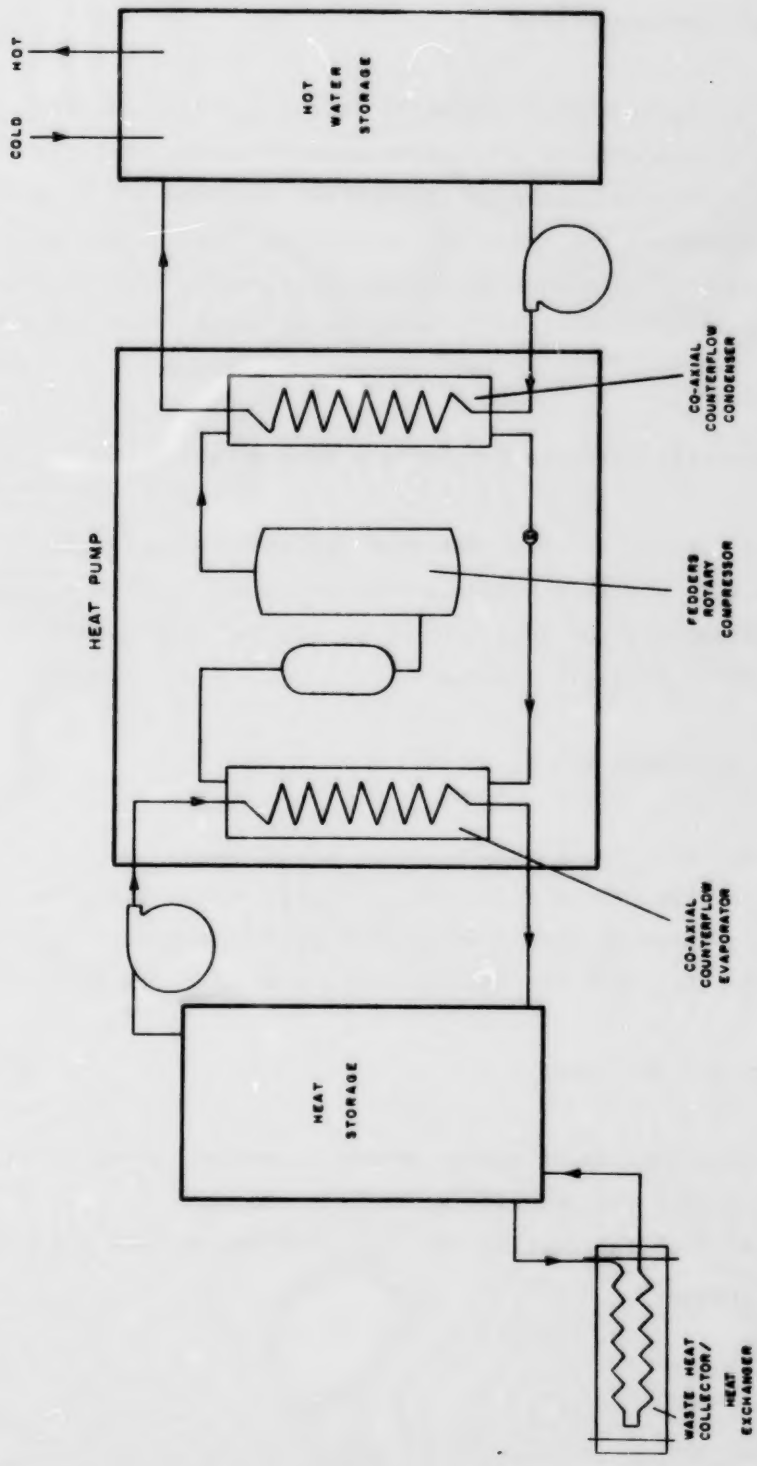


Fig. - 4 - WASTE HEAT RECOVERY WITH HEAT PUMP

1.2.1 Economic Considerations

Decisions involving capital investments in the dry cleaning and laundering service sector do not always promote energy conservation. Capital funds for energy conservation systems must compete with other ventures and therefore the industry requires an equal or better rate of return on investments for such projects. Thus, waste heat recovered systems must perform exceptionally well, require the least space for installation, and offer competitive returns on investment.

1.2.2 Site Specific Criteria for Service Demonstration

The successful outcome of this heat pump application is largely determined by site specific criteria and the selection of a suitable heat pump. The approach to the selection of a site for trials reflects this importance. These criteria include:

(i) Management Attitudes and Capabilities

- o Are they fully supportive of the application?
- o What are their attitudes to energy conservation?
- o Is there an energy management system in place?
- o Do they have the operational and maintenance capabilities?

(ii) System Cost and Performance

- o Relationship of energy demand to system output profile affects the need for waste heat storage;
- o Ease of integrating the heat pump system into existing system.

(iii) System Maintenance

- o Availability of qualified service personel.

(iv) Cost Savings

- o Type and local cost of energy supply;
- o Equipment use cycles.

1.2.3 Site Selection

Betty Brite, a Canada wide operation of franchised dry cleaning establishments, was selected for this study. The Betty Brite plant at 218 Laurier Street East in Ottawa operates a dry cleaning and laundery service using conventional equipment. This is an established business, with a solid clientele and is managed by Mr. J.G. Brennan, a certified stationary engineer and Vice-President of Betty Brite. Betty Brite is fully supportive of the application and has instituted commendable maintenance and operating procedures in support of energy conservation. Energy utilization records have been maintained for the plant throughout the past ten years and laundery services are operated continuously from 07:00 to 13:00 hours, six days per week. Duty cycles can be managed with minimum thermal storage and space limitations are typical of the plants visited by the project team.

1.2.4 Heat Pump Selection

For this application, the project team has selected the Fedders Carnot II Solar Compression Furnace. A product brochure is included in Appendix A. The ability of this unit to operate efficiently at high evaporating temperatures equips it for a wide variety of heat recovery applications. Where heating or cooling of processed liquids is required, such liquids can be pumped directly through the evaporator or condensor which are both

equipped with cupro-nickel inner tubes. Brines, degreasing solutions and a variety of mildly corrosive process liquids can be handled directly without the losses associated with intermediate heat exchangers.

The Fedders unit is available in four capacities from 20,000 to 80,000 BTUs per hour (nominal capacity) and is supplied complete, ready to operate when water and power lines are connected. The unit is equipped with a rotary compressor with internal overloads and counterflow coaxial condensors and evaporators. Automatic thermal expansion valves are factory installed. Contactors, relays and other components are factory wired into the control compartment and the entire assembly rests on a heavy gauge steel base with structural support members housed in a baked enamel steel cabinet. The Fedders unit is considered state-of-the-art because of its packaged design and use of the rotary compressor which accounts for its remarkable operating efficiency i.e., significantly higher coefficient of Performance (COP) compared to typical reciprocating compressor heat pumps.

1.3 PROJECT APPROACH

The project has been carried out in two phases:

- o Phase A - Design Development and Economic Evaluation of the Heat Pump Based Waste Heat Recovery System
- o Phase B - Service Demonstration.

1.3.1 Phase A Objectives

The primary objectives of Phase A are:

- o identify the optimum size and performance of the heat pump installation;

- o select equipment and prepare an engineering design for the proposed application which will ensure successful implementation meeting all design and regulatory codes and requirements;

- o prepare a statement of economic viability, capital cost, operating cost, benefits and payback as perceived by the plant operator;

- o prepare a statement of energy savings for this plant and estimate its effect on a nationwide basis;

- o recognizing that readily available and proven equipment is to be used to permit early implementation, review the research and development requirements which may yield further benefits e.g. heat pump working fluids, heat exchanger technology, materials technology, instrumentation and control technology;

- o determine the need for any new operating or maintenance skills and its impact upon operational management.

1.3.2 Phase B Objectives

The primary objectives of Phase B are:

- o to confirm the sizing and performance of the heat pump installation as a result of the service demonstration;

- o confirm the economic viability, capital cost, operating cost, benefits and paybacks as perceived by the plant operator during the actual service demonstration;

- o confirm and evaluate the potential energy savings for this plant and its effect on a nationwide basis based on actual service records and experience using the waste heat recovery system;

- o evaluate the need for operating or maintenance skills and its impact upon operational management based on actual experiences obtain during the performance of the demonstration.

1.3.3 Phase A - Design Development and Economic Evaluation

Phase A was performed under the following specific tasks:

Task 01 - Establishment of Operational and Design Criteria

The project team audited the operation of the service demonstration site to determine the following:

- o laundery equipment operating cycles

- o flow rate and temperature of waste water

- o flow rate and temperature of delivery hot water to laundery equipment

- o control parameters governing delivery of hot water to laundery equipment and waste water dumps

- o space limitations for installation of waste heat recovery systems

- o site modifications to facilitate installation of waste heat recovery system e.g. power supply, boiler plumbing, steam delivery piping

- o requirements for backup system operation on failure of waste heat recovery system

Task 02 - System Design and Integration

The project team performed a preliminary engineering design of the waste heat recovery system. This involved:

- o concept development

- o control logic development

- o load analysis

- o component sizing and selection (heat pump, storage units, piping, optional heat exchangers, maintenance facilities)

- o component optimization by computer simulation using TRNSYS

- o system integration

- o selection of suitable system controls.

Task 03 - Preliminary Cost Benefit Analysis

The project team performed a preliminary cost benefit analysis of the optimized waste heat recovery system since the incentive to implement such systems in this service sector are contingent on better than average returns on investment. Because the primary goal of the project is to stimulate the exploitation of heat pumps in Canada, it will be required

that the economic benefits of the proposed application be attractive to the industry. This condition is subject to no major incentive programs which might be offered by the public sector. The project team determined the life cycle costs of the proposed system and compared these to the existing system. The analysis includes both quantitative and qualitative factors with the objective of determining the overall feasibility of the proposed system, firstly as a sound investment for industry and secondly in full support of the goals of the National Energy Program.

1.3.4 Phase B - Service Demonstration

Phase B of the project involved the procurement of material, assembly and installation of the waste heat recovery system. The demonstration system was equipped with a data acquisition unit to monitor all performance parameters including fluid flow rates, temperatures and power requirements. The service demonstration would be expected to operate for a period of six months, during which time reliability and maintenance factors could be evaluated.

Monthly progress reports submitted to the NRC project scientist would detail all maintenance actions scheduled or unscheduled, monthly service and inspection reports, energy demands, load requirements and system performance of the proposed waste heat recovery system and energy demand by end use and backup system utilization. On completion of the service demonstration, JFH would update the cost benefit analysis using actual data derived from the service demonstration. The service demonstration would continue for a period of two years under the sponsorship of Betty Brite and at no cost to the Crown. Betty Brite would continue to submit monthly progress reports to the NRC project scientist, until the 31st of March of 1986. The intent of Betty Brite is to allow the NRC to continue to track the performance of the heat pump waste heat recovery system after completion of the service demonstration which was scheduled for the 31st of March, 1984.

Phase B would be performed under the following tasks:

Task 01 - Procurement of Material

JFH would procure all material required for the proposed heat pump waste heat recovery system and data acquisition system to remain resident with the installed unit.

Task 02 - System Installation and Predemonstration Trials

JFH would install the proposed waste heat recovery system and perform a sequence of predemonstration trials to verify the operation of the system inclusive of resident data acquisition system. In addition to the resident data acquisition system, an additional data acquisition system capable of continuously monitoring all performance functions of the heat pump waste heat recovery system inclusive of the industrial capacity washing machines would be installed.

Task 03 - Service Demonstration

The service demonstration would involve the in-service use of the as-installed heat pump waste heat recovery system for a period of six months.

Task 04 - Data Analysis and Synthesis

JFH would review all data recorded by the data acquisition system and compare these with the results obtained from the TRNSYS computer simulation used in Phase A. System running time, power consumption, system coefficient of performance, backup system utilization and plant energy demand by end use would be assessed. Any discrepancies between system prediction and system performance as measured by the continuous monitoring data acquisition system would be investigated and reported. JFH would compare the economic analysis of Phase A with data obtained from the service demonstration.

Task 05 - Reporting

JFH would submit monthly project reports to the NRC project scientist. Each report details all work performed during the reporting period, including:

- o all maintenance actions
- o monthly service and inspection reports
- o data registered by the data acquisition system
- o energy demand by end use, load requirements, system utilization and system COP
- o backup system utilization.

JFH would further update the economic benefit analysis using data obtained from the service demonstration.

Task 06 - 24-month In Service Reporting - Service Demonstration Extension

This final task does not form an integral part of Phase B - Service Demonstration, but represented a commitment by Betty Brite to continue to submit reports to NRC, thereby allowing NRC to continue tracking maintenance and operating performance of the heat pump waste heat recovery system for a period of two years after completion of the service demonstration. Betty Brite would submit data from the resident data acquisition system to NRC in a format mutually agreed upon between NRC and Betty Brite. JFH would assist in the preparation of a format consistant with the reporting procedures of Task 05.

2.0 OPERATIONAL AND DESIGN CRITERIA

2.1 TEST SITE SELECTION

The project team inspected four dry cleaning establishments in the Ottawa area for possible candidates as the heat pump test site. The following general conditions were found:

- Make-up water for the boiler is used to generate low pressure steam for direct use in pressing equipment;
- Steam is passed through a splitter valve with cold tap water to generate hot water of about 70°C;
- The hot water is used in industrial capacity washing machines;
- Waste water containing dissolved soaps, soils, lint, etc. is dumped periodically at approximately 60°C;

Since the heat pump system design would be based on site specific criteria, test data from a particular dry cleaning installation would be required. The desired test location should:

- representative of typical plants;
- have management which is supportive of energy conservation in general and the heat pump trials in particular;
- have records of past energy consumption.

All operators interviewed were agreeable to taking part in the project, however, Betty Brite (a Canada wide operation of franchised dry cleaning establishments) best satisfied our criteria. The Betty Brite plant is located at 218 Laurier Ave. E., Ottawa, Ontario.

2.2 TEST SITE OPERATING CONDITIONS

In order to obtain data for operation simulation programs and design analysis, the washing machine at Betty Brite Cleaners was monitored during September, 1983 for:

- hot and cold water feed rates (direct reading inline flow meters);
- drain time;
- inlet water temperature profile (thermocouples soldered to the copper pipes);
- drain water temperature profile;
- duty cycles (hour meters);
- machine cycle starting point (recorded by store personnel).

Actual testing was subcontracted to: Hovey & Associates
2378 Holly Lane
Ottawa, Ontario
K1V 7P1

Table 1 summarizes typical washing machine water usage data. Table 2 indicates the frequency of occurence of start operations in the wash cycle. Table 3 identifies the daily duty cycle.

TABLE 1: TYPICAL WASH CYCLE

			FIRST WASH	SECOND WASH	THIRD WASH	FIRST RINSE	SECOND RINSE	THIRD RINSE
FILL	DURATION (MINUTES)	HOT	2.5	2.5	2.0	2.0	2.0	---
		COLD	---	---	---	---	5.6	6.0
	FLOW kg/hr	HOT	1362	1362	1144	1090	1035	---
		COLD	---	---	---	---	21.8	24.5
	VOLUME litres	HOT	56.8	56.8	38.2	36.3	34.5	---
		COLD	---	---	---	---	20.4	24.5
	TEMP (°C)	HOT	60	60	60	60	57	---
		COLD	---	---	---	---	21	19
DRAIN	DURATION (MINUTES)		0.5	0.5	0.5	0.5	1.0	0.5
	TEMP (°C)		51	50	50	46	43	19

TABLE 2: TYPICAL START CYCLE

Number of loads started at:

Wash 1:	9
Wash 2:	17
Wash 3:	4
Rinse 1:	0
Rinse 2:	15
Rinse 3:	5
Starch	3
	53 loads over 7 days

TABLE 3: TYPICAL DUTY CYCLE

DATE	MACHINE TIME (hours)
Wednesday	4.0
Thursday	4.4
Friday	5.3
Saturday	2.25
Monday	5.6
Tuesday	5.3
Wednesday	5.55
7 days	32.5 hours

Total store hours: 7 days X 5.5 hours/day = 38.5 hours
Duty Cycle = 32.5/38.5 = 0.84

2.3 TEST DATA ANALYSIS

Test results from Betty Brite showed that waste water of 830 litres/day at 50°C was produced by a standard industrial washing machine. This value is depressed for the following reasons:

a) Hot water production was recently changed from steam heating to two conventional electric hot water tanks (total volume 454 litres). This modification was put in place by Betty Brite to reduce dependency on their boiler. However the hot water tanks do not have cycle time sufficient to to meet the Betty Brite hot water demand. The results are:

- o hot water temperatures are lower;
- o many washes are being done in cold water;
- o the full machine cycle is rarely used;
- o clothing is being shipped to other plants.

All of these factors contributed to a reduced waste water production.

b) Business in September is traditionally slow, so a higher duty cycle is expected during other months.

2.4 DESIGN PARAMETERS

It was decided that the initial heat pump system design would be based on the following parameters.

- o Flow - 454 litres/hr @ 100% duty cycle

- o Wash Inlet Temperature - 60°C
- o Wash Drain Temperature - 50°C
- o Water delivery control as encountered during testing.

The parameters selected are typical for shirt laundering facilities encountered in Canadian establishments.

The economic analysis will be performed for duty cycles of 50%, 75% and 100%.

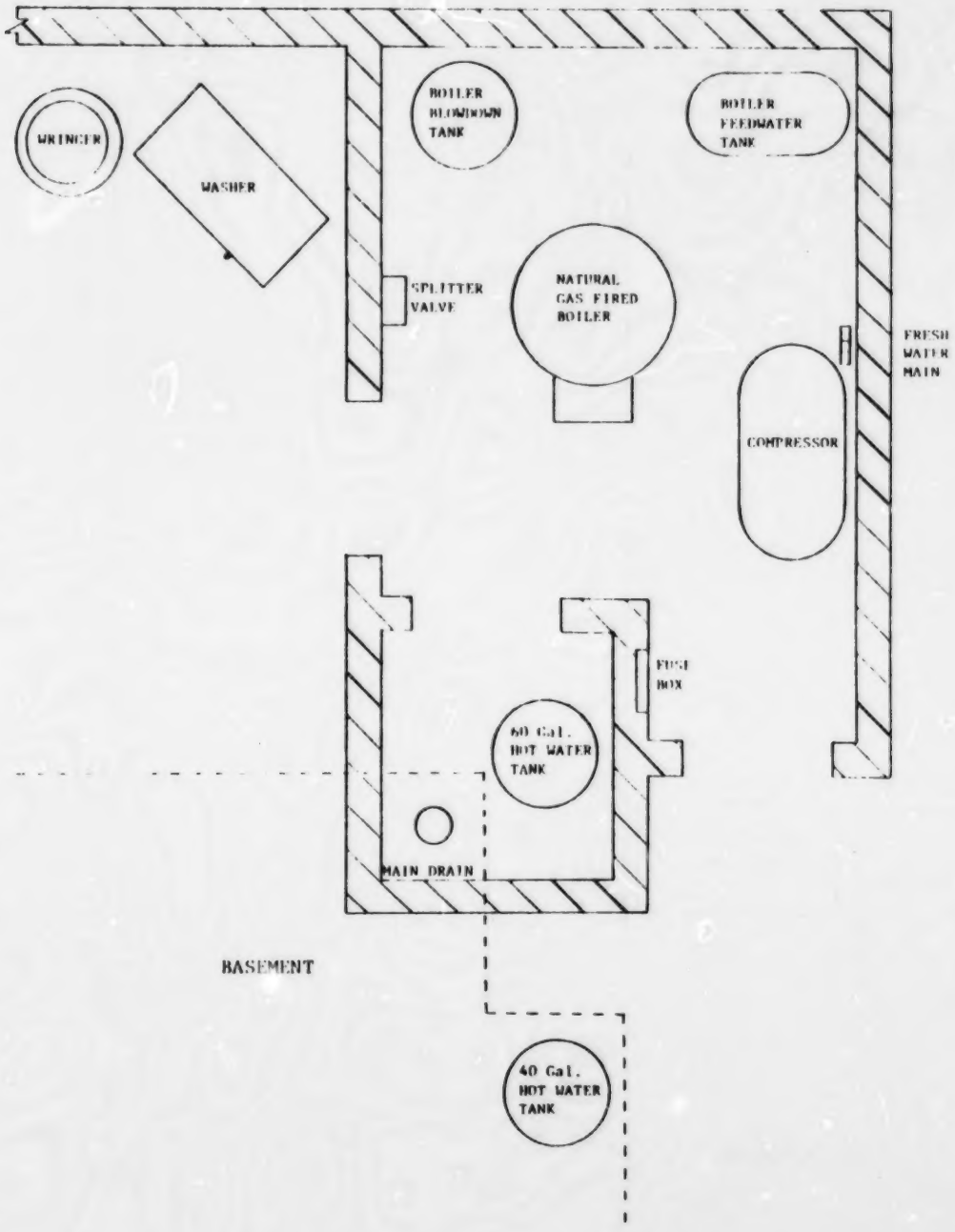
Transient response of the heat pump system computer simulation will be driven by inputs based on actual washing machine control performance.

2.5 EQUIPMENT INSTALLATION

Figure 5 shows the present Betty Brite Washer and Boiler Room layout.

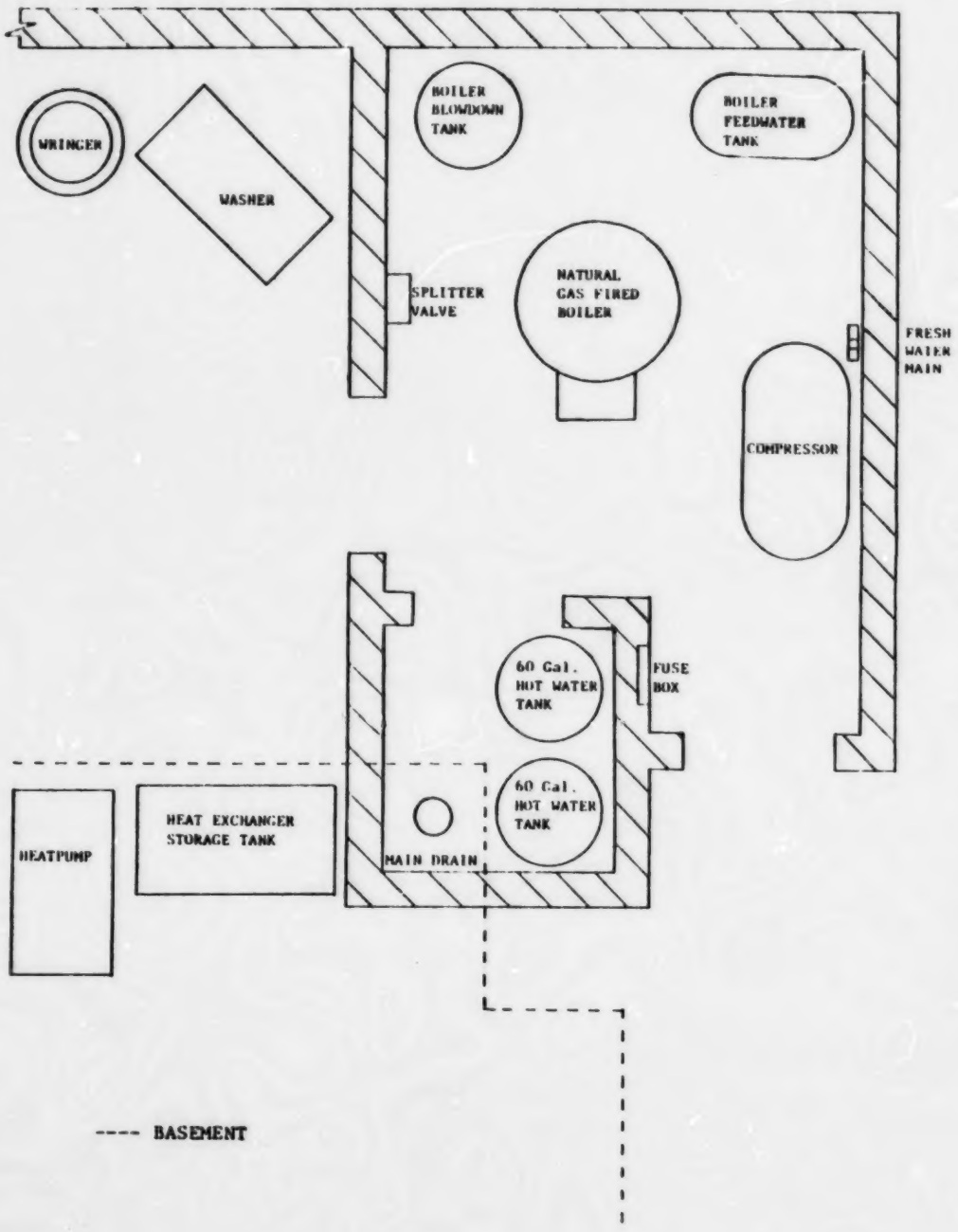
The most suitable location for the heat pump system is in the basement as shown in Figure 6. The second hot water tank would be installed next to the present tank upstairs. This setup permits the easiest installation, operation, monitoring and servicing of the equipment.

FIGURE 5



PRESENT EQUIPMENT SITE INSTALLATION

FIGURE 6



PROPOSED EQUIPMENT SITE INSTALLATION

3.0 SYSTEM DESIGN AND INTEGRATION

3.1 GENERAL

The project team carried out preliminary engineering design of the proposed waste heat recovery system. This involved:

- o concept development;
- o control logic development;
- o load analysis;
- o component sizing and selection (heat pump, storage units, piping, optional heat exchangers, maintenance facilities);
- o component optimization by computer simulation using the TRNSYS models;
- o system integration;
- o selection of suitable system controls.

The equipment used for the waste heat recovery system will meet all design and regulatory code requirements for Canada. The Fedders water to water heat pump commonly known as the Solar Compression Furnace is UL approved and currently available in the Canadian market. Furthermore recognizing that readily available off-the-shelf equipment could be used, the project team concentrated its efforts on component sizing and selection of off-the-shelf items in an attempt to keep the total capital cost as low as possible. This reflects the desire that the proposed waste heat recovery system be cost effective for this service sector.

3.2 CONCEPT DEVELOPMENT

Concept development of the waste heat recovery system is predicated on trial results obtained under Task 01 and design goals established by the project team to facilitate penetration of the system into the dry cleaning and laundering service sector.

Preliminary testing revealed that the proposed system must heat city water (main) from 18`C to 60`C at a rate of approximately 454 litres/hour. Therefore the system must be designed such that it is capable of supplying approximately 80,000 kJ/hr of heat energy. Four methods are presently deemed available to do this:

- o heat pump;
- o heat pump plus heat exchanger;
- o heat pump plus auxilliary heater;
- o heat pump plus heat exchanger plus auxilliary heater.

The project team selected the fourth option for system modelling. This fourth option covers all the methods of reclaiming and generating heat for the proposed system. The project team thus could use one concept with the TRNSYS computer code and optimize component size and selection. The component specifications are as follows:

- o heat pump system - minimum heating capability of 80,000 kJ/hr;

- o heat exchanger - minimum flow rate of 454 litres/hr with a minimum efficiency of 40%;

- o auxilliary heater - capable of supplying 80,000 kJ/hr should the heat pump primary system fail (electric resistance heating on back-up).

3.3 COMPUTER SIMULATION PROGRAM DEVELOPMENT

The computer code used is the TRNSYS application program. This program was developed by the University of Wisconsin and is available through time sharing facilities with the National Research Council. The program was orginally designed to model solar heating systems, but because of its modular design, it can be used for a multitude of applications.

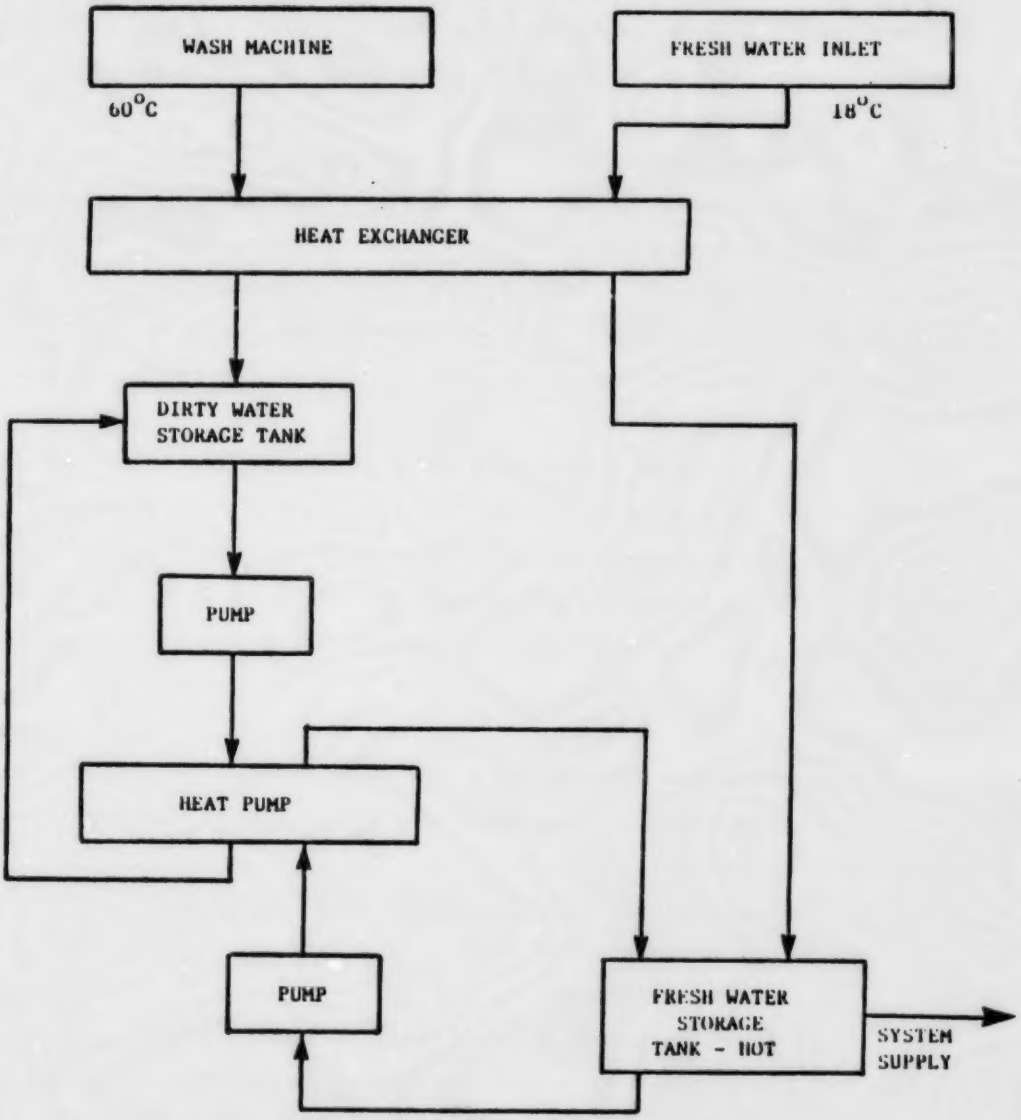
The approach taken was to define the requirements from the simulation and then run a cross-check to determine if the TRNSYS modules met these specific needs. Upon reviewing the functions of the modules, it was found that the heat pump, heat exchanger and storage modules were not fully satifactory and therefore had to be improved for use with the TRNSYS simulation. Basic program descriptions are included in Appendix B.

The waste heat recovery system currently simulated is structured as shown in Figure 7. The system uses constant recirculation through the waste water storage tank thereby reclaiming as much heat as possible from the waste water. The programs have been designed to be able to output any calculated variable at the users discretion. For the current system the outputs are:

- heat transferred through the heat exchanger, kJ/hr;
- flow rate of city water through the heat exchanger, litres/hr;
- heat transferred through the heat pump, kJ/hr;
- power consumed by heat pump, kJ/hr;
- COP of heat pump;
- power consumed by auxilliary pumps, kJ/hr;
- heat added by auxilliary heater on demand, kJ/hr;
- power consumed by auxilliary heater, kJ/hr;
- total system COP;
- temperature of storage tanks, `C;
- system running time, hours;
- total power consumed by auxilliary heater, kJ/hr.

FIGURE 7

COMPUTER SIMULATION SYSTEM SCHEMATIC



The Fedders Carnot II Solar/Compression Furnace (heat pump) was specifically modelled for use in the TRNSYS computer code. Data received from the manufacturer relating to the performance of the heat pump is interpreted by a program developed by the project team.

A series of bench mark trials were carried out on the modified TRNSYS code. The modified program functioned properly.

3.4 LOAD ANALYSIS

A load analysis was performed using sample data obtained from the results of the Betty Brite Cleaners site investigation. Flow rates and temperatures obtained from the investigation were input into the TRNSYS simulation as required. Appendix C details the results of the TRNSYS simulations.

From the data it can be seen that the 80 series heat pump is best suited for a utilization of 100%. The 40 series heat pump was not capable of maintaining the hot water storage at the minimum required temperature without the use of an auxilliary heater. Both of these units have high flow rate demands through the evaporator and condenser coils. Lower flow rates through the evaporator and condenser coils were simulated, however, it was found that the coefficient of performance dropped and the system could not supply sufficient heat energy to maintain the storage tanks at the desired operating temperature. The system development was therefore based on the 80 series heat pump.

3.5 COMPONENT SELECTION

Component specifications are based on the results of the TRNSYS simulation. All material selected are off-the-shelf items. No component required specific design for the installation except for piping. Flow rates for the pumps were determined from the heat pump specification. The heat pump was selected entirely on the basis of its heating capacity. The heat exchanger was selected for its ability to operate under laundery conditions. Component specifications are identified in Figure 8. The final selection of all system components will be based on availability at time of trial system installation under Phase B of this project. However the estimated cost of the system was obtained for the economic analysis. The prices quoted are retail prices and would be expected to be lower for larger quantities. Figure 9 identifies the approximate costs of the components by major assembly.

3.6 SYSTEM INTEGRATION

The project team carried out additional site surveys of the Betty Brite plant in order to determine the optimum location for installation of a prototype waste heat recovery system proposed under Phase B of the project. These site surveys revealed that there was insufficient space in the boiler room to install both the waste heat recovery system and monitoring equipment. Figure 10 identifies present equipment installed in the boiler room. Betty Brite was willing to move certain pieces of equipment under the proposed Phase B of the project, but this would have created additional problems with the presently installed plumbing so it was decided to find another area in the plant where the test equipment could be located. After consultation with Betty Brite it was agreed to install the test equipment in the basement of the plant. The basement is presently used for storage and there is ample room to locate all the equipment and testing apparatus. Figures 11, 12, 13, and 14 identify the locations of major components and piping proposed for Phase B of the project. Some modification will be required for the present hot water feedline to the washing machine and of the present drain system as well, but the remainder of currently installed plumbing would be unchanged.

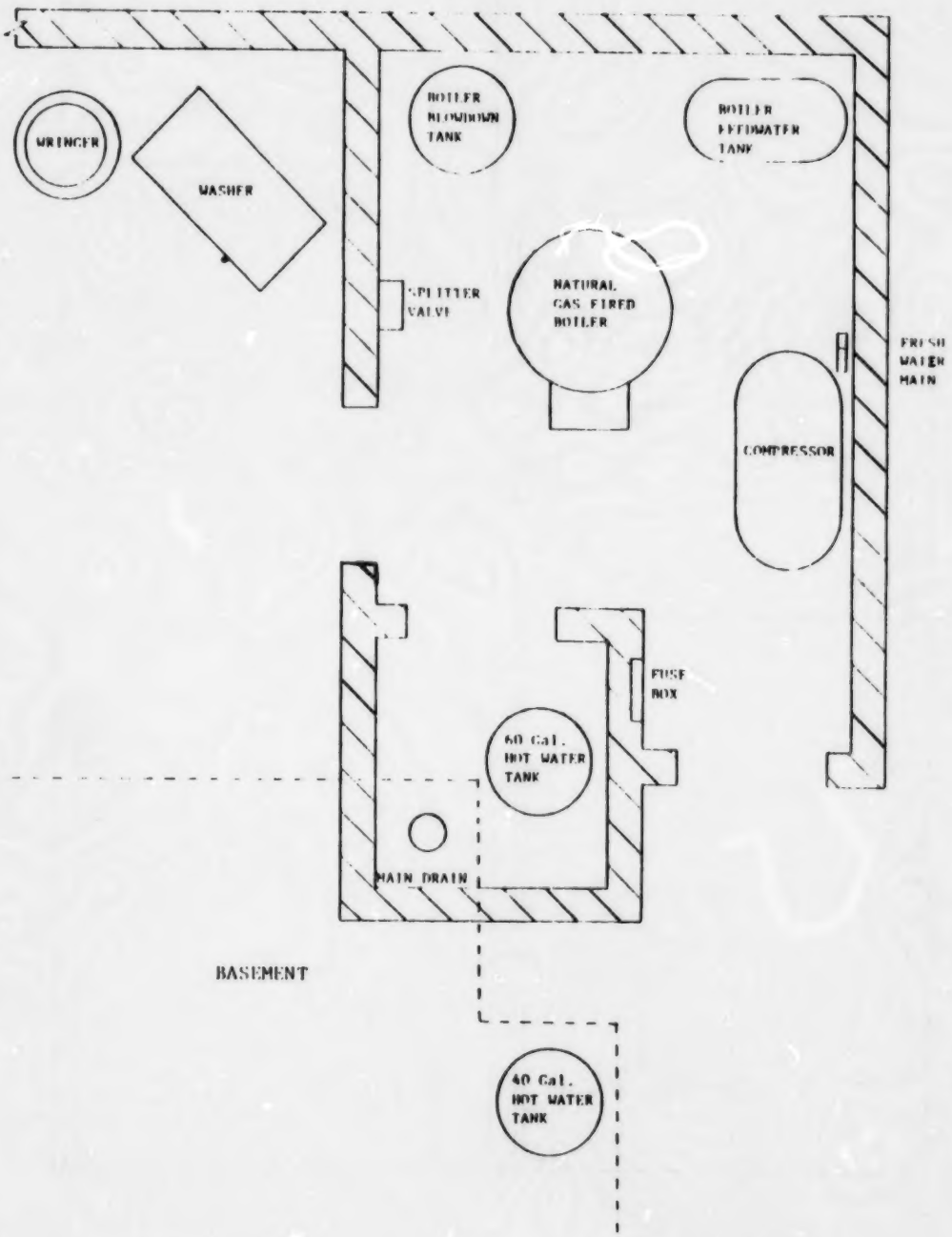
FIGURE 8: COMPONENT SPECIFICATIONS

ITEM	SPECIFICATION
Heat Pump	- 80,000 kJ/hr heat rate - water to water unit - self contained unit - high COP factors - corrosive resistance coils
Heat Exchanger	- 455 litres/hr minimum flow rate - corrosive resistant construction - readily serviceable
Axuilliary Pumps	- 154 litres/minute flow rate - capable of 60°C temperatures - corrosion reistant construction - 517 kpa operating pressures
Storage Tank (waste water)	- 545 litre capacity - capable of containing a heat exchanger - corrosion resistant
Storage Tank (fresh water)	- 545 litre capacity - self contained auxilliary heater of 80,000 kJ/hr capacity

FIGURE 9: COST OF SYSTEM COMPONENTS

ITEM	COST
Heat Pump - 40 series	$2,900
60 series	$4,175
80 series	$5,330
Heat Exchanger	750
Pumps	250
Plumbing and Fittings	200
Storage Tanks - waste water	$500
fresh water	$330 - with integral auxiliary heaters
Installation	$500

FIGURE 10



PRESENT EQUIPMENT SITE INSTALLATION

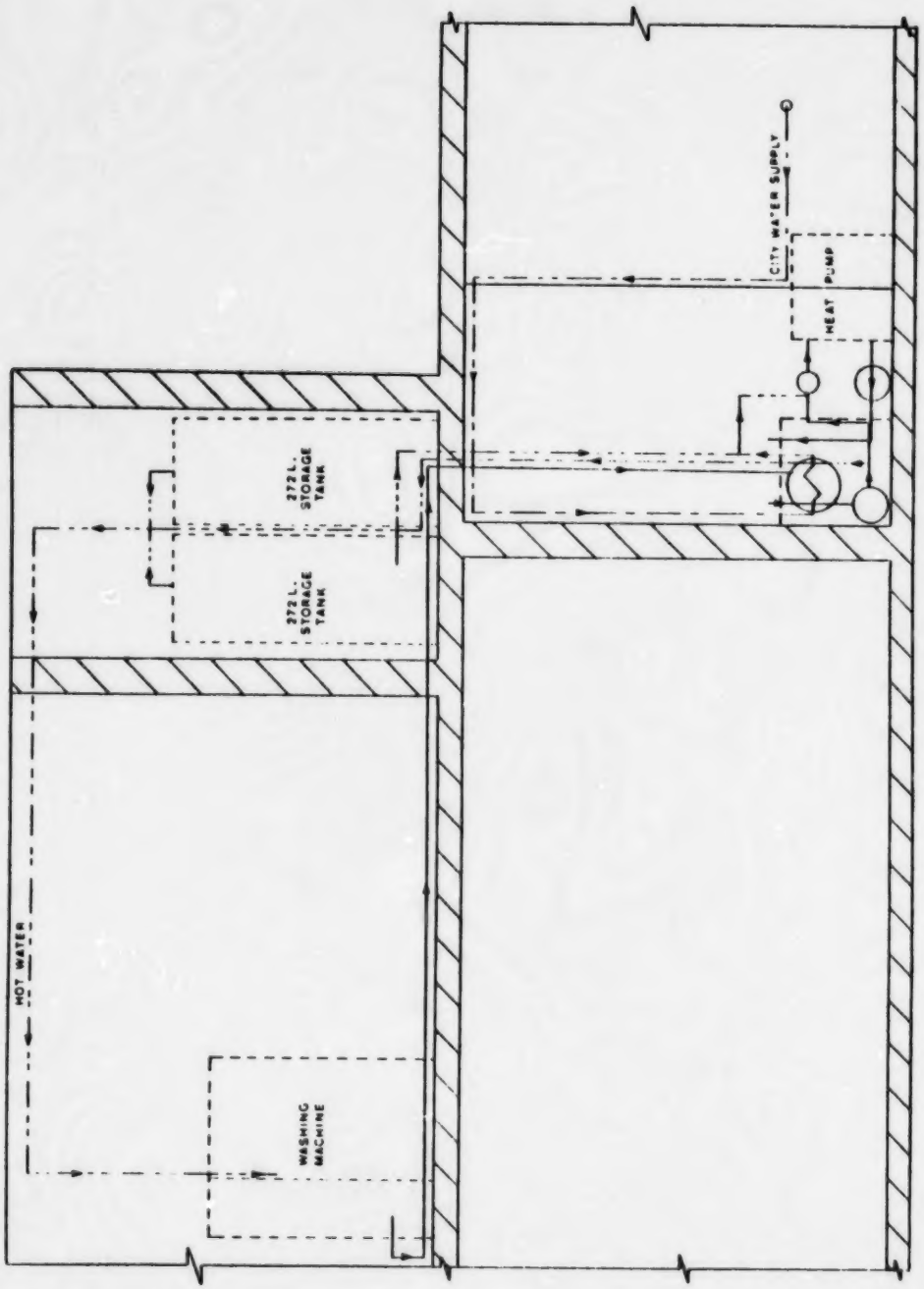
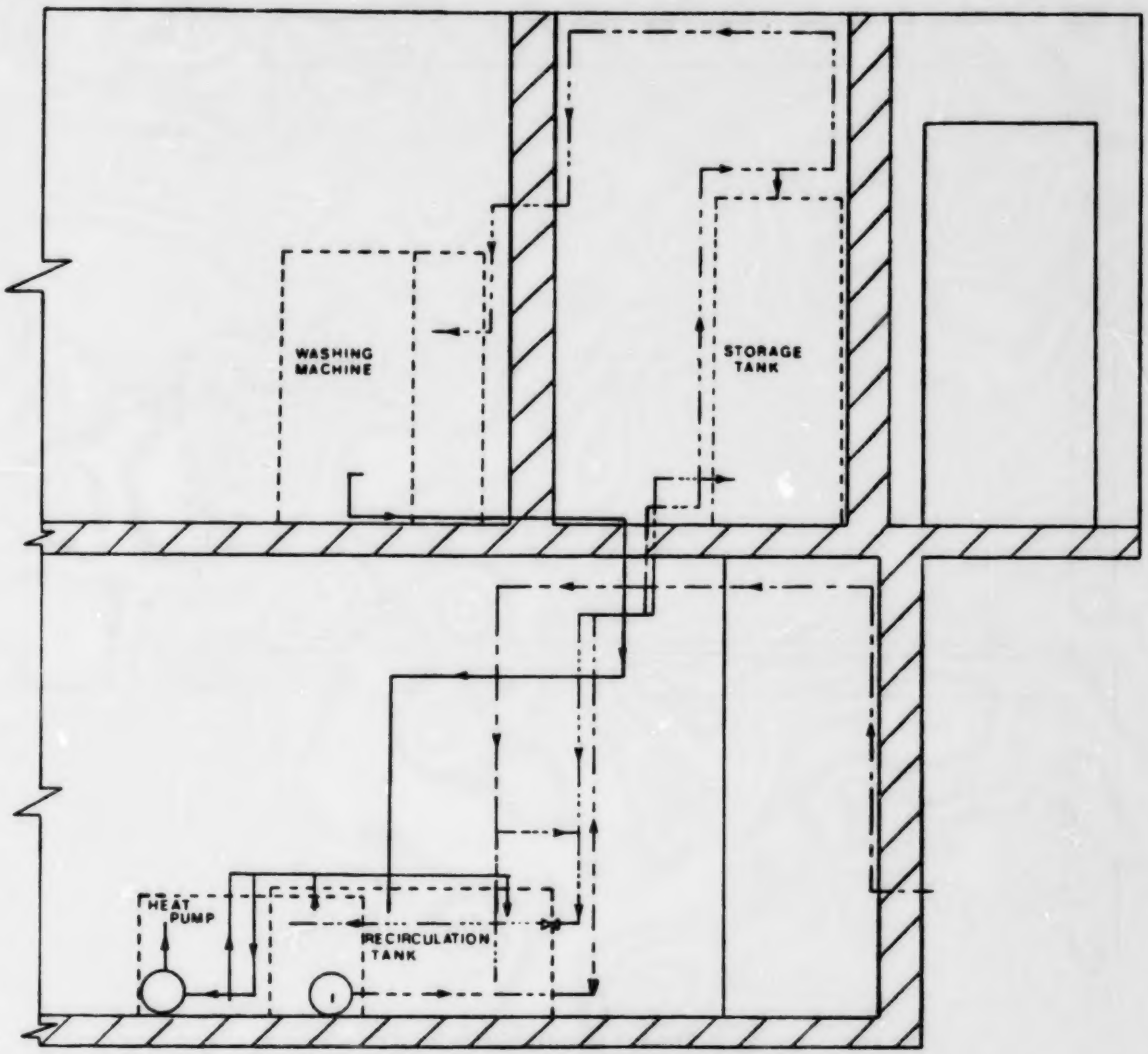


FIGURE 11

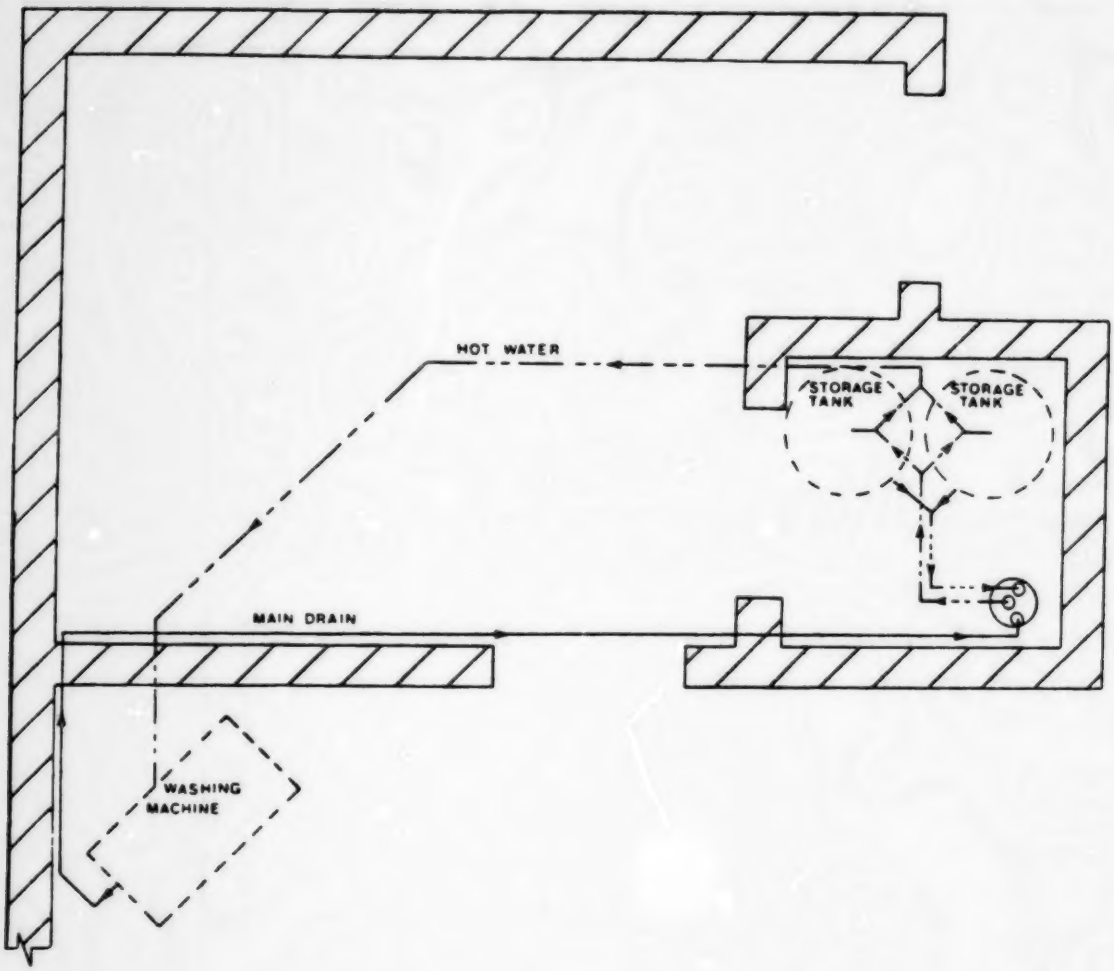
FRONT VIEW OF PROPOSED PIPING LAYOUT

FIGURE 12



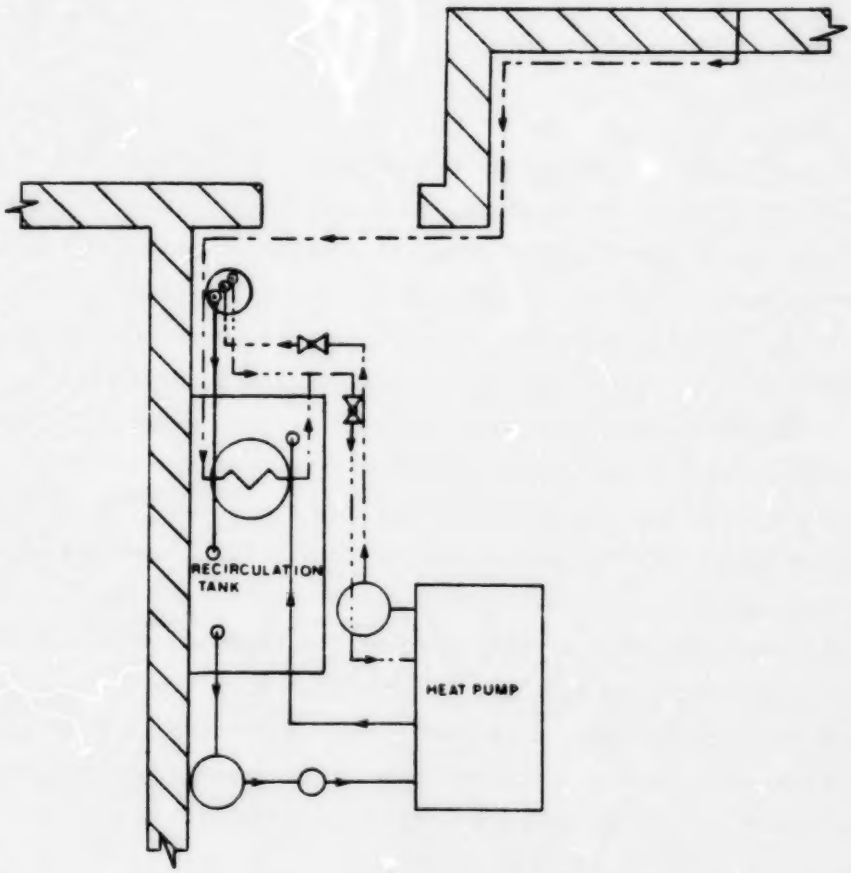
SIDE VIEW OF PROPOSED PIPING LAYOUT

FIGURE 13



PLAN VIEW OF PROPOSED
BOILER ROOM PIPING LAYOUT

FIGURE 14



PLAN VIEW OF PROPOSED
BASEMENT PIPING LAYOUT

4.0 PRELIMINARY COST BENEFIT ANALYSIS

4.1 GENERAL

The application of heat pump technology to waste heat recovery is receiving greater attention as an energy conservation measure. Heat rejected from industrial processes is efficiently recovered by the system for use as process heat; thus overall energy utilization is increased over conventional heat delivery systems. However, the increased efficiency increases the capital cost of the total system because of the need for system modifications, additional heat exhangers, related piping, controls and the heat pump. Although the capital cost per unit of delivered thermal energy is greater for the proposed system than for conventional electric resistance or fossil fueled hot water heating, the system operating and maintenance costs are lower. For economic viability, the O&M cost advantage of the proposed system must generate sufficient revenue to offset the higher capital cost, over time. This condition applies to add-on heat recovery systems as proposed for this application. However, for the case of new plant construction, part of the added capital cost may be offset by eliminating the present boiler as the primary source of process heat.

4.2 ECONOMIC MODEL

The cost effectiveness of the proposed waste heat recovery system is determined by the market value of the useful products the system generates (i.e. hot water) and by the cost of owning and operating the system. In this study, the conventional or reference system is an oil-fired boiler with splitter valve typically used in dry cleaning and laundering plants. The break-even capital cost (BECC), which is the cost that should not be exceeded if a venture is to be profitable, was selected as the basis for comparing the waste heat recovery system to the conventional system.

The BECC was selected for the comparison for the following reasons:

- o the approach is similar to that used by industry for making investment decisions;
- o the resulting equation is general and can be used for analyzing the waste heat recovery system against any reference system, comparing each on the same basis;
- o prior knowledge of the proposed system's actual capital cost is not required. The total capital cost is obtained separately and can be frequently updated to reflect changes as the detailed design evolves. These changes will not affect the BECC value.

The BECC is derived from the venture-worth method which determine the value of a venture over and above a minimum required return on investment. The value of capital investment that sets the venture-worth equal to zero is the BECC. The generalized BECC economic model is derived in Appendix D.

It is key to the understanding of the results of the proposed economic model that one realizes that investments in the private sector are based on after-tax dollars rather than pre-tax dollars as is commonly used in the public sector. Thus the BECC is particularly conservative when compared to other public sector studies on the cost benefits of heat pump systems.

Revenue from the expected reduction in energy costs cannot be used in whole towards the purchase of capital goods. This revenue is directly reproducible as "bottom-line" profit and taxed at the applicable corporate rate. Thus any venture must use after-tax dollars and returns on investment must be significantly higher.

Discussions with owner/operator's in the dry cleaning and laundering service sector make reference to the available profit dollar for purchase of new equipment. For any such system as that proposed here, the return on investment must be particularly attractive to the owner/operator's. JFH and NRC recognize this situation and thus the economic analysis is fully based on the owner/operator's perception of the economics of introducing heat pumps into their operation.

To fully substantiate the model under these conditions, JFH has selected rates of return which are commonly quoted by sector representatives for 1983.

In addition, Canadian taxation policy for wholly-owned Canadian companies presently operating in the Province of Ontario is used for the numerical solution of the generalized economic model. Thus, the following parameters apply:

corporate tax rate:	25%	(t)
depreciation rate:	20%	(d_r)

investment tax credit: 20% (TCI)

high risk rate of return: 20% (i_m)

discount rate: 15% (i)

depreciation period: 5 years (r)

system life: 15 years (n)

The major factor which will affect the continued economic viability of the proposed heat pump system is energy pricing. Revenue to pay for the capital cost of the system is primarily derived from energy cost savings (exclusive of maintenance savings and capital replacement funds). Thus it is essential that a realistic conventional energy price scenario be utilized. Growth rates predicted by the Bureau of Managment Consulting (BMC) - Conventional Energy Price and Demand Scenarios, BMC, 1980 are used for the economic model.

They are, for Ontario:

fuel oil (commercial user)

1983-1985 : 9.6%
1986-1997 : 1.7%

electricity (commercial user)

1983-1985 : 3.3%
1986-1997 : 0.7%

Base year energy costs are:

fuel oil : $0.323/litre (1982)
electricity : $0.0426/kWhr (1982)

It must be noted that these costs are average energy costs inclusive of quantity discounts.

To complete the selection of appropriate parameters, we refer to the revenue term, R_K. We find that only COP and U remain to be selected. From the results of TASK 02, system modelling confirms that peak system COP is 3.5 Since it is desirable to ascertain the effect of COP on BECC, the economic model makes use of four possible system COP's; 3.0, 3.5, 4.0 and 4.5. Thus, the benefits which might be derived from significant technical improvements to heat pump design can be assessed under expected increases in system COP. Utilization, U, is varied as 50%, 75% and 100%. The selected utilizations also correspond to rated nominal capacities of each of the three models of heat pumps offered by Fedders, namely 40,000 Btuh (50%), 60,000 Btuh (75%) and 80,000 Btuh (100%) units.

The effect of public sector incentives to industry in the form of taxable grants must be determined. At present, an $800 taxable grant is available for the installation of heat pumps. In those test cases where the taxable grant is used, the total $800 taxable grant is revenued in the first year, R_1.

The economic model is also used to determine the benefits accrued from electric resistance heating (i.e. domestic hot water). System COP for electric resistance heaters without heat recovery lies between 0.9 and 1.0. The capital cost of such a system is estimated to be about $500 (1982). Thus, the introduction of a heat pump waste heat recovery system can be compared to that of electric resistance heating. Accumulated net revenue for the proposed waste heat recovery system and electric resistance heating (COP = 0.9) are shown in Figures 15 to 17.

4.3 ECONOMIC MODEL - RESULTS

The economic model developed under Section 2.0 for the proposed waste heat recovery system has been used to generate the Tables and Figures that follow. Breakeven capital cost as a function of payback period have been determined for various scenarios involving COP and U (Table 4 to 7 and Figure 18 to 20 refer). Matching these results with the estimated capital costs of the proposed system the respective payback periods of Table 8 are obtained.

4.4 ECONOMIC VIABILITY

The proposed waste heat recovery system is fully feasible from an investment viewpoint. Payback periods of about three years are considered by the industry to be very good for high risk ventures especially when seven years is considered the norm by industry representatives.

Accumulated revenue generated by the proposed waste heat recovery system results in net gains over electric resistance heating after seven years of operation at 50% utilization, six years at 75% utilization and five years at 100% utilization.

The economic analysis is based on the perspective of the owner/operator and thus cannot be directly compared to public sector analyses which are unaffected by taxation policies.

FIGURE 15

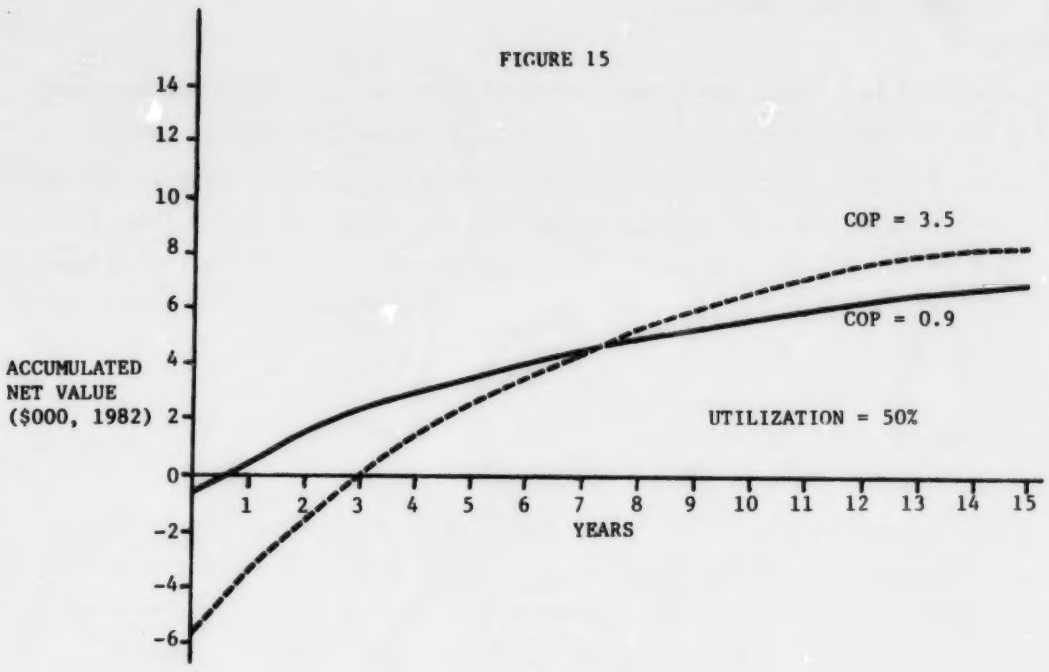


FIGURE 16

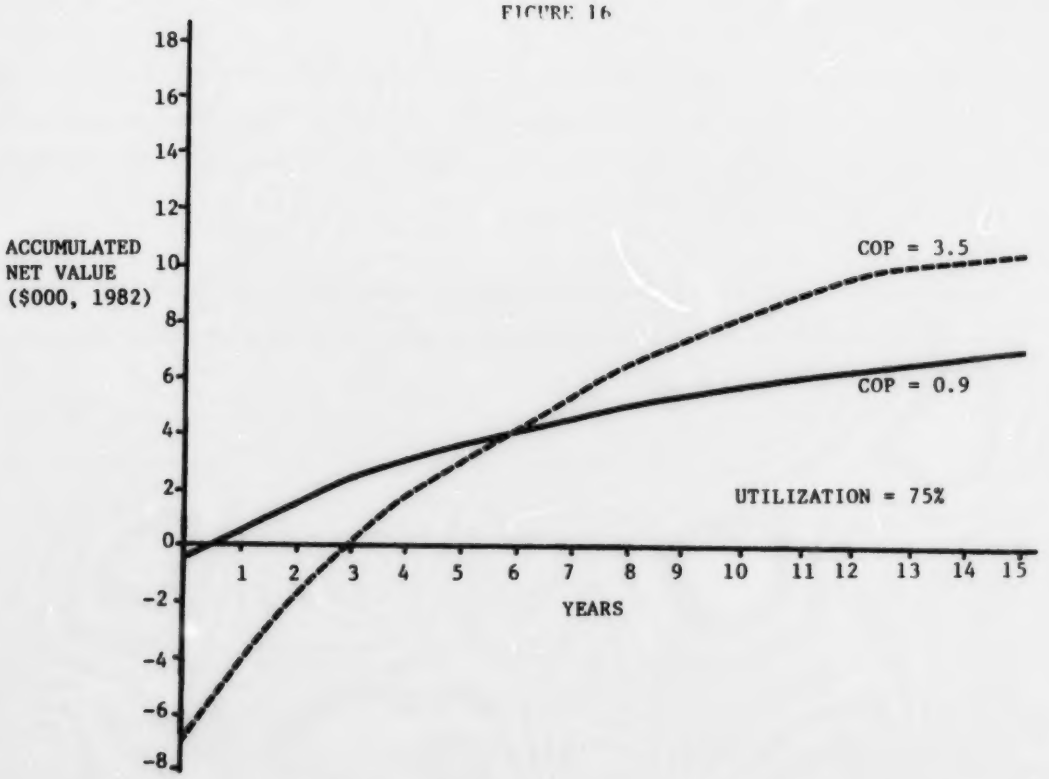


FIGURE 17

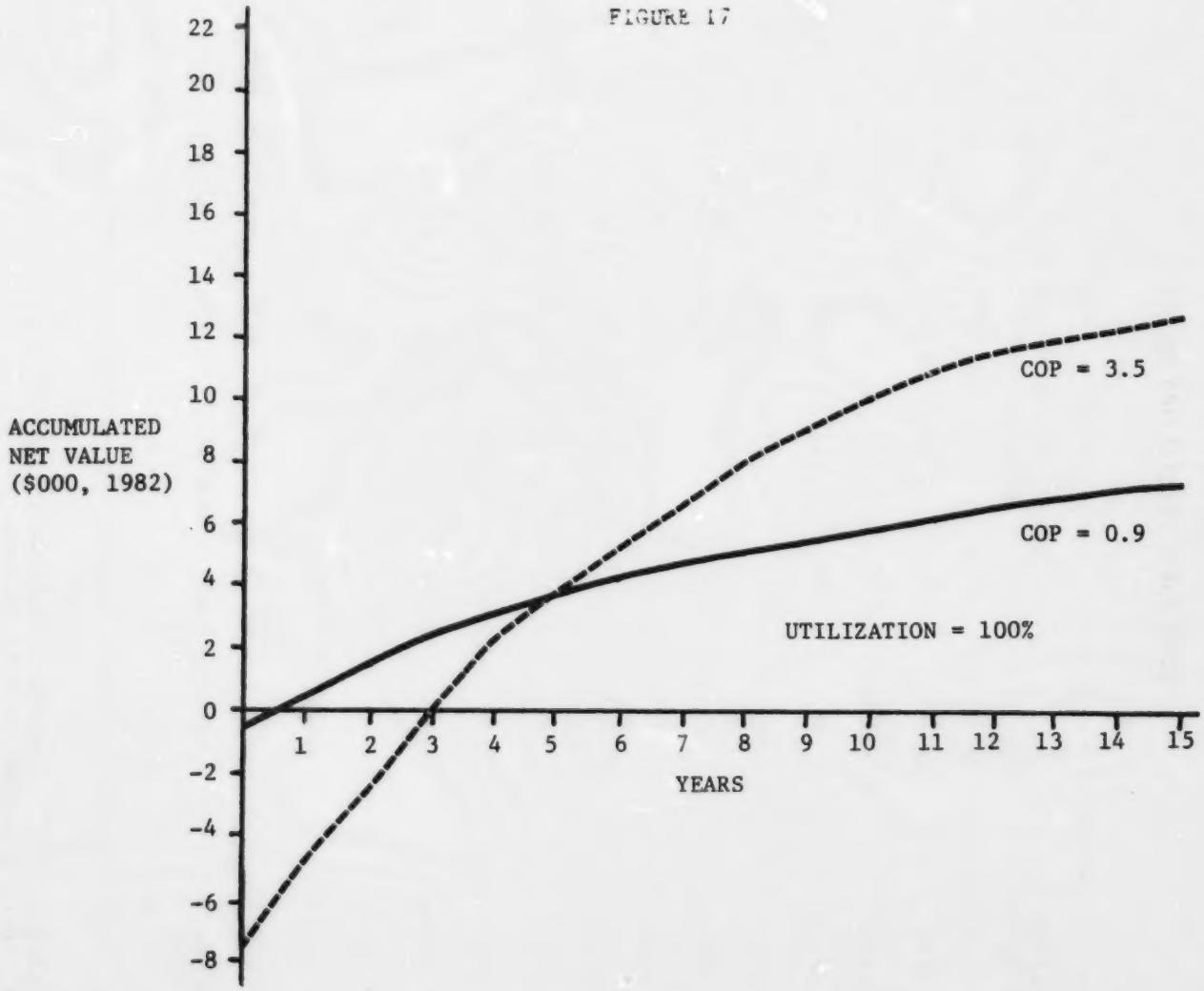

22
20
18
16
14
12
10
8
6
4
2
0
-2
-4
-6
-8
ACCUMULATED
NET VALUE
($000, 1982)
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
YEARS
COP = 3.5
COP = 0.9
UTILIZATION = 100%

TABLE 4:

BECC FOR A UTILIZATION OF 50%

PAYBACK PERIOD OF:	COEFFICIENT OF PERFORMANCE OF: 0.9	1.0	3.0	3.5	4.0	4.5
1 year	1,047	1,178	1,968	2,025	2,067	2,100
2 years	2,023	2,273	3,773	3,880	3,961	4,023
3 years	2,938	3,294	5,431	5,584	5,699	5,788
4 years	3,610	4,053	6,707	6,897	7,039	7,150
5 years	4,202	4,720	7,827	8,049	8,216	8,345
6 years	4,722	5,306	8,810	9,060	9,248	9,394
7 years	5,180	5,827	9,673	9,948	10,154	10,315
8 years	5,583	6,275	10,431	10,727	10,950	11,123
9 years	5,937	6,674	11,096	11,411	11,648	11,833
10 years	6,249	7,025	11,680	12,012	12,261	12,455
11 years	6,524	7,333	12,192	12,539	12,799	13,002
12 years	6,765	7,605	12,643	13,002	13,272	13,482
13 years	6,978	7,844	13,038	13,409	13,687	13,904
14 years	7,165	8,054	13,385	13,766	14,052	14,274
15 years	7,331	8,239	13,690	14,080	14,372	14,599

(Excludes $800 tax credit available in year 1)
All costs are shown in 1982 dollars.

TABLE 5:

BECC FOR A UTILIZATION OF 75%

PAYBACK PERIOD OF:	COEFFICIENT OF PERFORMANCE OF: 0.9	1.0	3.0	3.5	4.0	4.5
1 year	1,040	1,238	2,423	2,507	2,571	2,620
2 years	2,044	2,419	4,669	4,829	4,950	5,043
3 years	3,015	3,549	6,756	6,985	7,157	7,290
4 years	3,676	4,339	8,321	8,606	8,819	8,985
5 years	4,260	5,037	9,698	10,031	10,281	10,475
6 years	4,777	5,653	10,909	11,285	11,566	11,785
7 years	5,235	6,197	11,974	12,387	12,696	12,937
8 years	5,639	6,678	12,911	13,356	13,691	13,950
9 years	5,998	7,103	13,376	14,209	14,565	14,841
10 years	6,315	7,479	14,461	14,960	15,334	15,625
11 years	6,596	7,811	15,099	15,619	16,010	16,314
12 years	6,845	8,104	15,661	16,200	16,605	16,920
13 years	7,065	8,363	16,155	16,711	17,129	17,453
14 years	7,260	8,593	16,590	17,161	17,589	17,922
15 years	7,433	8,795	16,972	17,556	17,994	18,335

(Excludes $800 tax credit available in year 1)
Note: inclusion of the $800 results in an additional $500/entry.
All costs are shown in 1982 dollars.

TABLE 6:

BECC FOR A UTILIZATION OF 100%

PAYBACK PERIOD OF:	COEFFICIENT OF PERFORMANCE OF: 0.9	1.0	3.0	3.5	4.0	4.5
1 year	1,033	1,297	2,877	2,990	3,075	3,140
2 years	2,065	2,565	5,565	5,779	5,939	6,065
3 years	3,093	3,805	8,080	8,385	8,614	8,793
4 years	3,741	4,626	9,935	10,314	10,599	10,820
5 years	4,318	5,354	11,569	12,013	12,346	12,605
6 years	4,832	6,000	13,008	13,509	13,884	14,176
7 years	5,289	6,573	14,276	14,826	15,238	15,559
8 years	5,696	7,081	15,392	15,986	16,431	16,777
9 years	6,058	7,533	16,376	17,008	17,481	17,850
10 years	6,381	7,933	17,242	17,907	18,406	18,793
11 years	6,669	8,288	18,006	18,700	19,221	19,626
12 years	6,924	8,603	18,679	19,398	19,938	20,358
13 years	7,152	8,883	19,271	20,013	20,569	21,003
14 years	7,354	9,131	19,794	20,555	21,126	21,571
15 years	7,534	9,356	20,254	21,033	21,617	22,071

(Excludes $800 tax credit available in year 1)
All costs are shown in 1982 dollars.

TABLE 7:

BECC for COP = 3.5 inclusive of Federal $800 taxable grant with i = 15% and r = 5 years

PAYBACK PERIOD OF:	UTILIZATION OF: 50%	75%	100%
1 year	2,595	3,037	3,478
2 years	4,300	5,134	5,967
3 years	5,796	6,978	8,158
4 years	7,110	8,599	10,088
5 years	8,261	10,024	11,786

(Includes $800 tax credit available in year 1)
All costs are shown in 1982 dollars.

TABLE 8:

PAYBACK PERIOD FOR WASTE HEAT RECOVERY SYSTEM

Utilization % (imp gal/hr)	capital cost installed ($1982)	system COP	Payback period (years)	
			WO/grant	W/grant
50	5430	3.5	3	2.75
75	6705	3.5	3	2.75
100	7860	3.5	3	2.75

FIGURE 18

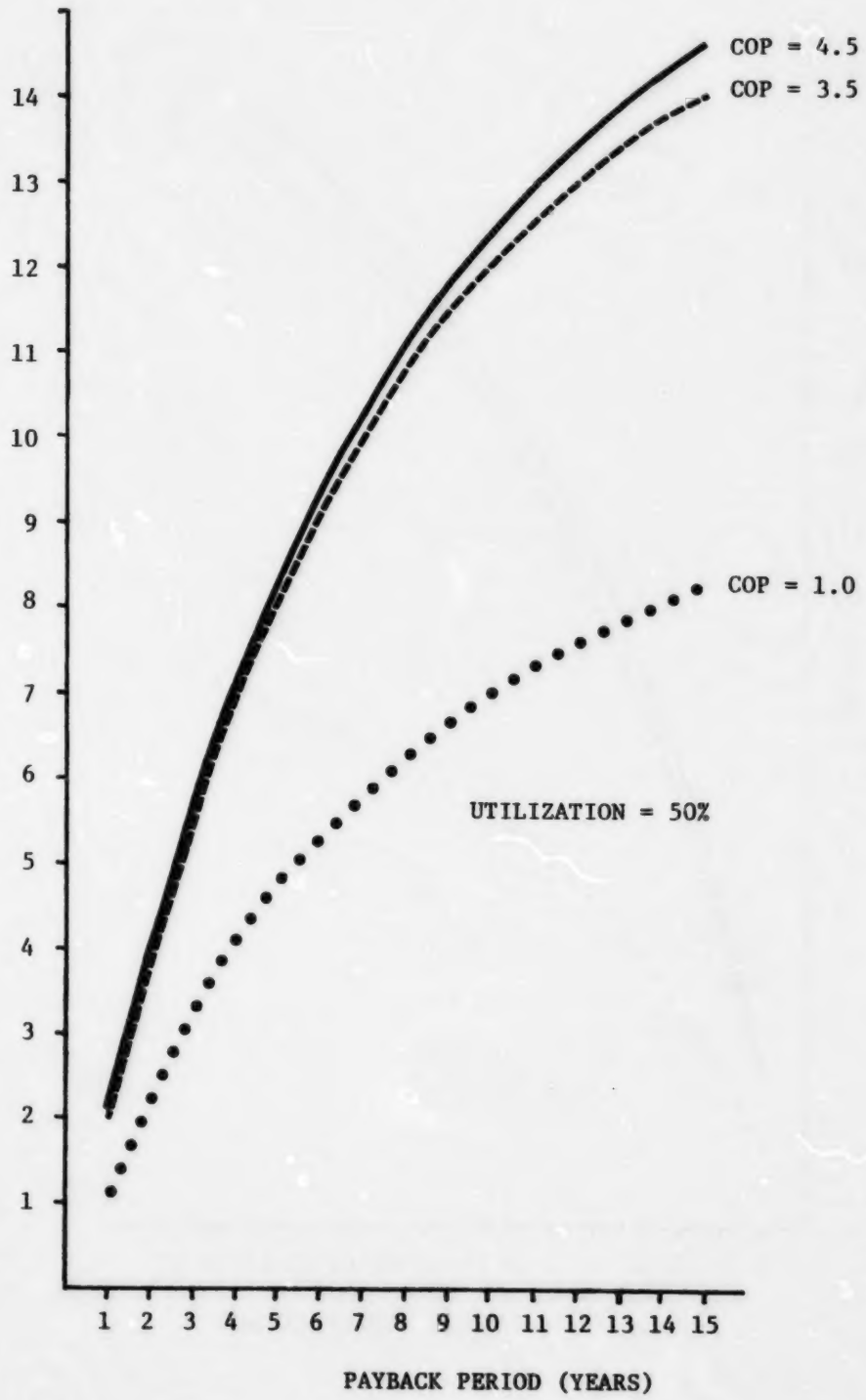

COP = 4.5
COP = 3.5
COP = 1.0
UTILIZATION = 50%
14
13
12
11
10
9
8
7
6
5
4
3
2
1
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
PAYBACK PERIOD (YEARS)

FIGURE 19

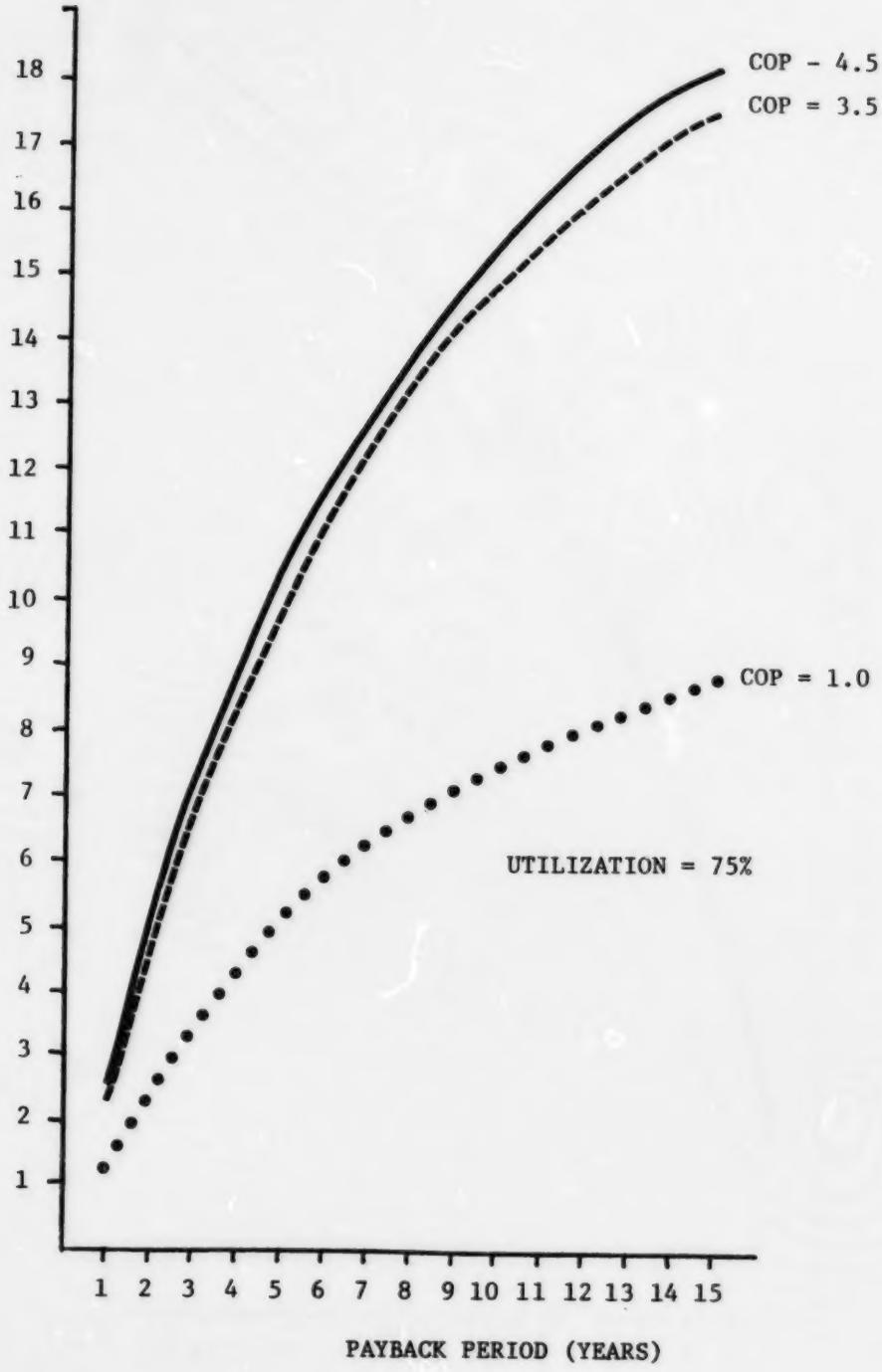
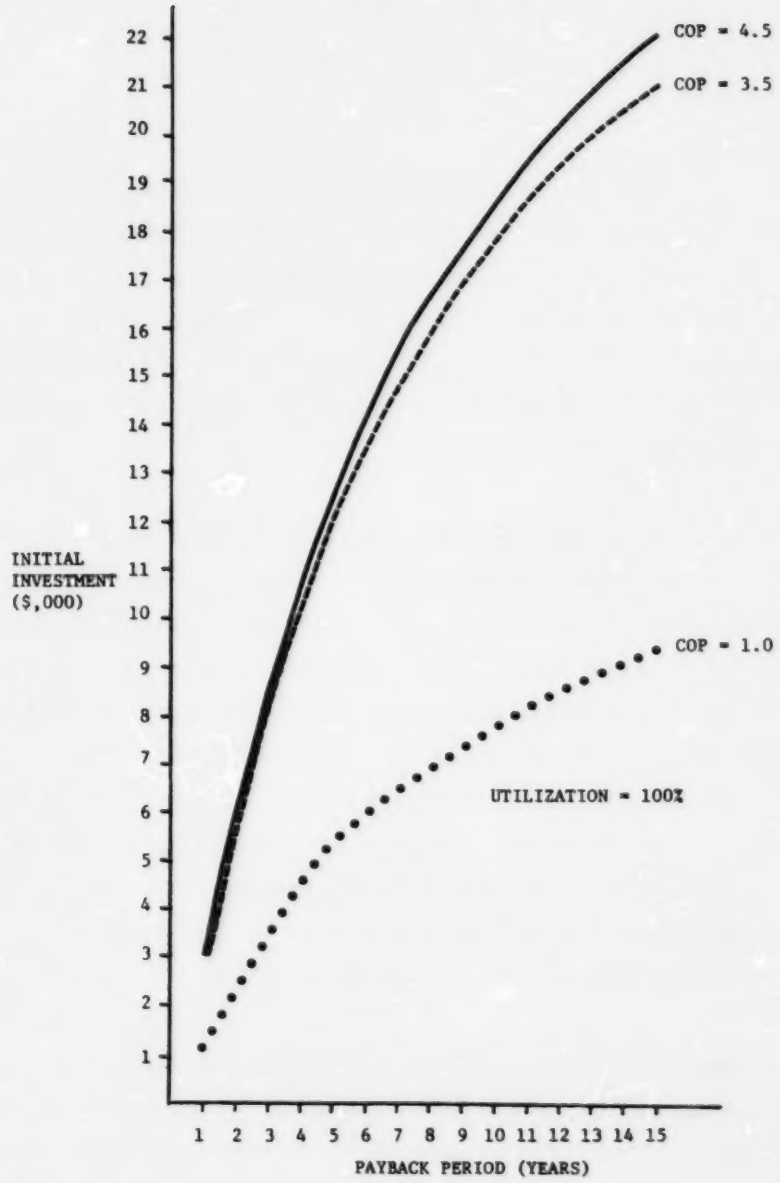

18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
COP - 4.5
COP = 3.5
COP = 1.0
UTILIZATION = 75%
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
PAYBACK PERIOD (YEARS)

FIGURE 20


COP = 4.5
COP = 3.5
COP = 1.0
UTILIZATION = 100%
INITIAL INVESTMENT ($,000)
22
21
20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
PAYBACK PERIOD (YEARS)

5.0 SERVICE SECTOR ENERGY DEMAND SCENARIOS

5.1 APPROACH

Currently there is very little information available on energy usage in the commercial sector as a whole. Although there are some estimates available of energy demand for the commercial sector the reliability of these estimates is uncertain. These estimates do not disaggregate demand into different commercial enterprises and hence they are of limited use in estimating the potential for heat pump waste heat recovery system penetration.

There are available, however, detailed estimates of the amount of energy of each fuel type consumed by 22 commercial enterprises. These estimates were compiled by the structural analysis division of Statistics Canada in 1978, using 1971 data. They are the output of an input-output based energy analysis model which measures the energy intensiveness of the goods and services produced in the Canadian I/O system. The numbers produced and the methodology used are published in the report "Energy Availability, Detailed Disposition, and Industrial Demand Co-efficients for Canada, 1971", by Deachman and Hamilton, August 1978. The results of this analysis form the basis for most of the data presented here on the dry cleaning and laundering service sector.

The commercial data which has been compiled for the base year 1980 is a projection of the 1971 I/O estimates. The approach for this projection involved a comparison of the 1971 I/O estimates with the 1971 Detailed Energy Supply and Demand publication to determine the relationship of each commercial enterprise demand to total demand. The 1971 S&D publication was then compared to the EMR model projected 1980 commercial demand and this was used to modify the 1971 I/O estimates as follows:

o Total Energy Consumption

Since total commercial consumption increased 20% from 1971 to 1980 it was assumed that each individual commercial enterprise would increase its consumption by 20%.

o Fuel Oil Consumption

Total commercial fuel oil consumption dropped 36% during the 9 years, thus each enterprise's fuel oil consumption was reduced by this percentage.

o Gas and Electricity Consumption

Gas plus electricity went up 283 X 10^{15}J in 9 years with gas forming 45% of the increase. Thus, the increase in each enterprise's combined gas plus electricity consumption was calculated based on assumptions noted above and this increase was prorated as 45% gas.

The results for the base 1980 are shown as follows:

Laundery and dry cleaning establishments account for 1.4% of the total commercial energy demand. By fuel type oil required

6.0 X 10^{15}J, gas 1.1 X 10^{15}J, and electricity 5.8 X 10^{15}J.

In projecting 1990 and 2000 commercial demand the same approach was used as that used to project the 1971 estimate to 1980. Specifically it was assumed that all enterprises would have the same percentage increase in total energy usage and decrease in fuel oil consumption. The increase in individual enterprises, gas plus electricity consumption, was prorated according to the overall increase. In addition, the consumption of other

fuels was prorated according to each enterprises total energy consumption. Thus, the demand for energy in the laundry and dry cleaning service sector became 4.5 X 10^{15}J for oil, 1.8 X 10^{15}J for gas, and 8.5 X 10^{15}J for electricity, and for the year 2000: 3.5 X 10^{15}J for oil, 2.5 X 10^{15}J for gas and 10.4 X 10^{15}J for electricity. In addition for the year 2000 it has been estimated that approximately 0.7 X 10^{15}J may be attributable to alternate fuels other than oil, gas, or electricity. These other fuels could include propane, solar heating, or other renewable and non-renewable sources.

5.2 ENERGY DEMAND BY END USE

The end use data is divided into the following four categories.

1. Steam: the heat energy used to convert water at 100°C to steam, and to heat this steam to the required temperature.

2. Hot Water (50-100°C): the heat energy used to heat water from 50°C to the required temperature, including preheating water for steam.

3. Hot Water (less than 50°C): the heat energy used to heat water up to a maximum of 50°C, includes preheating water for use at more than 50°C and for steam.

4. Direct Heat: all other heat energy.

The breakdown of industrial consumption by end use was estimated for 20 manufacturing industries by General Energy Associates in the United States. The basis for this work is a very detailed industrial data base developed from a survey at 140 plants situated in the U.S.

To estimate Canadian energy usage, GEA was supplied with data from the 1976 survey of manufacturing energy purchases, plus data on numbers of employees and manufacturing sites. By comparing these Canadian data against the U.S. database an estimate was made of Canadian industrial consumption by end use.

It was found that steam represented 44.1% of the total energy demand in all manufacturing sectors. Hot water from 50-100°C involved 5.7% of the total energy demand. Hot water from 0-50°C represented 9.3% of the total energy demand. While direct heat represented 40.1% of the total energy demand.

The results of this work indicated that heating water from 0-50°C does not consume a very significant amount of the total energy demand for industrial sectors. Since these water heating applications are the only applications in the manufacturing sector which offer any significant potential for currently feasible heat pump technologies, it is clear that these technologies can not be expected to displace a large fraction of the conventional fuels used for heat in the manufacturing sector.

In projecting the growth in industrial energy consumption by fuel type, it was assumed that fuel consumption would change in proportion to the overall change in industrial usage of that fuel.

The projection of the end use distribution had to take into account the heat content of the various fuels. In the base year, it was assumed that electricity provided no heat content. To retain this assumption for 1990 and year 2000 it appeared to be inconsistent with the demand projections which predict a 20 year increase of only 26% for total coal, oil, and gas consumption. At the same time a 153% increase is predicted for electricity plus other fuels. For the non-fossil fuels to grow at such a quick rate they would likely have to be used for some of the growth in the industrial use of heat energy. It was therefore assumed that one half of the growth in electricity and other fuels would provide heat energy.

Thus, industrial energy demand by end use projected for the year 1990 were 44.1% for steam, 5.7% for hot water from 50-100'C, 9.3% for hot water from 0-50'C, and 40.9% direct heat. Projected industrial energy demand by end use for the year 2000, resulted in the following: 44% for steam, 5.7% for hot water from 50-100'C, 9.3% hot water from 0-50'C, and for direct heat 41%.

5.3 DRY CLEANING AND LAUNDERING SERVICE SECTOR

At the present time very little information is available regarding the split of energy consumption within the dry cleaning and laundering service sector by end use. Proper instrumentation of a typical dry cleaning and laundering plant will provide a reasonable estimate of energy demand by end use. Under Phase B of this project, the Betty Brite plant selected for the prototype installation of the waste heat recovery system would be instrumented to provide a breakdown of the end use energy demand. Steam is used for pressing equipments and has been extensively used throughout the industry to generate hot water through splitter valves. Hot water from 50-100'C represents the primary demand for hot water in the laundry and dry cleaning service sector. Hot water less than 50'C is comparable to domestic hot water demands for such plants. The total demand for water at temperatures less than 50'C are comparatively low compared to the demand for hot water in excess of 50'C. Demand for direct heat is limited purely to space heating within the plants, lighting, fans, exhaust systems, and powering pressing, washing machine, and dry cleaning equipment. For the purpose of this study, JFH examined the energy demand records of the Betty Brite plant at 218 Laurier Street and has estimated that approximately 50% of the total energy demand for the plant was required to support the laundering operations of the plant.

In order to estimate the reduction in total energy demand to a plant given that the heat pump waste heat recovery system can operate at a simulated level of system COP of 3.5 it is estimated that the

introduction of a waste heat recovery system will result in a 70% reduction in energy required to support the laundering operation. Since the total demand of the system is estimated to represent 100% thus we may assume that the introduction of waste heat recovery systems into this service sector can result in a 35% reduction in total energy demand for a plant.

Total energy demand by the service sector in 1980 is estimated at 12.9×10^{15}J, 15.0×10^{15}J for 1990, and 17.1×10^{15}J for year 2000.

The potential penetration of heat pump waste heat recovery systems into the service sector is unknown at this time. However, based on the economic model presented earlier it is estimated that heat pump waste heat recovery systems could attain a significant penetration. If it is assumed that by the year 1990 approximately 10% of all establishments had converted to waste heat recovery systems, then it is estimated that the total energy saving to the service sector would be 3.5% of the annual total energy demand, or 0.53×10^{15}J. It is furthermore estimated that by the year 2000 most plants presently in operation in the year 1980 will have been required to replace present heating and hot water systems. With the change effected towards waste heat recovery, it is possible that well in excess of 50% of the total service sector could be converted to heat pump waste heat recovery systems thereby resulting in a total service sector saving of 17% in annual energy demand. The 17% saving would result in a 2.91×10^{15}J energy saving for the total service sector. In comparing this energy saving to the estimated total industrial energy demand for the year 2000 for Canada the introduction of waste heat recovery systems based on heat pumps into the dry cleaning and laundering service sector could result in a 0.1% total reduction in energy demand for the industrial sector in Canada. This is a notable saving considering that it is estimated that the demand for energy in the laundry and dry cleaning service sector represents approximately 1.4% of the total commercial energy demand for Canada in the year 2000.

6.0 HEAT PUMP SERVICE DEMONSTRATION

6.1 PROTOTYPE SYSTEM DESIGN

6.1.1 System Design

The design of the test installation is to a great extent based on the system structure of the simulation model. The system, based on the use of a heat pump and storage tanks for waste water and feed water, is designed for use in small commercial laundery type applications.

6.1.2 Waste Water Storage Tank Design

The waste water storage tank is an important component in the waste heat recovery system. The design specifications of the waste water storage tank were dictated in part by the computer simulation model and the requirements of the Betty Brite installation. The design specifications for the waste water storage tank are as follows:

- o constant volume type
- o 590 litres capacity
- o corrosuve resistant construction
- o readily cleanable
- o insulated
- o covered

JFH assessed the availability of off-the-shelf waste storage tanks, but none could meet the required specifications. It was therefore decided to design and build the storage tank to meet the specifications of this particular application. Hovey & Associates Ltd. was selected to design and build the tank. Detailed drawings of the prototype tank are included in Appendix E.

The waste water storage tank consisted of a welded steel frame with an inner shell constructed of 304 stainless in order to resist corrosion. It was covered with two inches of fibreglass insulation. This was all covered by an outer steel shell. The tanks dimensions are 1 m x .71 m x 1.24 m and has a capacity of 590 litres. Ports are drilled into the tank to allow for bulkhead connections of the plumbing for the heat pump and drain. A shelf is attached to the side of the tank to support the circulating pumps and strainer. The controller is also attached to the side of the tank. The tank stands on a elevated platform to allow for gravity draining of the tank for cleaning purposes.

6.1.3 Auxilliary Component Selection

Once the major prototype components had been selected there remained the selection of the auxilliary equipment. In selecting the auxilliary components the specifications of the larger components must be known in order for the auxilliary components to match larger items such as the heat pump, waste storage tank and feedwater storage tanks. To complete the prototype system the auxilliary equipment items required are recirculation pumps, filter, piping, fittings and valves. Much of the component selection for the prototype system was based on equipment recommendations from Fedders. The auxilliary equipment suggested by Fedders is fully compatible with the heat pump and the storage tanks. Table 9 gives a list of the auxilliary components, specifications and supplier. All required equipment was obtained from local suppliers. Care was taken in the location of equipment for the prototype system to ensure compact packaging.

TABLE 9: AUXILIARY COMPONENTS

ITEM NO.	COMPONENT	BASIC SPECIFICATIONS	BRAND NAME	PART/ MODEL/ TYPE
1	Recirculation pumps 2 required	75 l/min flow rate @ 140 kPa stainless steel and aluminum construction 115 volts single phase motor Seals to withstand temperatures to 80`C	Grundfos	UPS 20-42F series
2	Ball valves 2 required	Rated working pressure of 830 kPa Water temperature of 80`C. Bronze 1 inch IPS	Watts regulator	#B326
3	Strainer 1 required	Cast iron 40 mesh screen, cleanable	Zurn	Z-1180
4	Diverter valves 4 required	1 inch IPS 3 way, brass	ENCO Standard	#55144
5	Temperature/ Pressure relief valve	Pressures to 830 kPa Temperatures to 80`C	Robertshaw	TP1100-014
6	Thermostat 2 required	Must be able to measure temperature in feedwater tanks in order to turn on the heat pump	Honeywell	T675A
7	Piping	Type L Copper tubing 1 inch nominal diameter		
8	Fittings	Copper fittings 1 inch nominal diameter		

6.2 PROTOTYPE SYSTEM INSTALLATION

6.2.1 Delivery of Components

A major delay was encountered in the project as the result of delays in the delivery of the heat pump by Fedders. Delivery of the heat pump was more than three months late and this held up the project because none of the design details could be finalized. JFH worked with specifications provided by Fedders for much of the design. As it turned out their specifications were wrong and the unit was actually twice as large as was claimed. Fortunately this did not cause major difficulties requiring a redesign of the system, but it did cause some problems during installation of the heat pump in the basement. Railings and doorframes had to be removed, and the heat pump just barely fit through the opening. All other components were delivered on time and there were no further delays.

6.2.2 Installation of Heat Pump Prototype System

Installation of the heat pump prototype system progressed rapidly once the heat pump was delivered. Hovey & Associates was responsible for the installation at Betty Brite Cleaners and designed and manufactured the appropriate support structure for the heat pump. Some parts of the installation were prebuilt so that as little time as possible was wasted. Work was performed only in the afternoons so that it did not disrupt the regular operation of the plant. The heat pump and waste water storage tank were both bolted to a U-channel frame and the plumbing connected. The hot water storage tank was located in a different corner of the basement from the original plans because the heat pump was larger and interferred with a door.

Temperature control is accomplished using Honeywell Model T-675AS controllers. There are two of them in the system. One is used for controlling the temperature of the hot water and the other is used in an

emergency shut-off when the waste water storage tank becomes too cold. This prevents the evaporator from frosting, causing the heat pump to trip its circuit breaker protector. They were originally installed on the plumbing leading to and from the heat pump, but after testing it was decided that this was not a suitable location for them and they were subsequently moved to another location to more accurately judge the average temperature of the system.

Another problem that arose after the installation was that the condensor side of the heat pump appeared to be hooked on backwards. During the installation the plumbers had followed the installation instructions supplied by Fedders, but on closer analysis of the heat pump it was discovered that this was in fact backwards and therefore had to be switched around. Photographs of the as-installed heat pump waste heat recovery system are shown in Appendix F.

6.2.3 Maintenance

In order to ensure that the heat pump waste heat recovery system operates at maximum efficiency, JFH determined the nature and frequency of maintenance that would be required on the system. Maintenance requirements were obtained by operating the system and inspecting frequently to determine what had to be cleaned and what time interval was required. A strainer was installed on the waste water side of the heat pump to trap lint and sediments. With the size of the strainer selected, it was found that it could last 14 days between cleaning. The waste water storage tank should be drained and rinsed once a month. The fresh water side of the system requires no maintenance. The heat pump should be descaled once a year to ensure optimum operating efficiency. Divertor valves were installed between the heat pump and the water pumps to facilitate this task. Descaling chemicals are added to the flexible lines and then circulated in a closed loop through the heat pump to clean the coils of the heat pump. The coils are then flushed and the system reconnected. The heat pump should also be serviced once a year by an authorized service representative in order to verify freon pressure.

6.3 DATA ACQUISITION SYSTEM

The data acquisition system used to monitor the performance of the heat pump prototype waste heat recovery system was based on the HP-IL system. It is used in conjunction with an HP-41C that allows the data to be recorded on microcassette. A sample of the data recorded during the operation of the prototype heat pump waste heat recovery system is listed in Appendix G. All data is available on IBM-compatible 5.25 inch floppy diskettes.

6.4 BETTY BRITE ENERGY END USE

As part of the service demonstration JFH performed an energy end use data survey of the Betty Brite Laundering and Dry Cleaning plant. The purpose of this monitoring was to help the National Research Council better assess the energy savings that are attributable to the heat pump waste heat recovery system. In order to accomplish this, Honey and Associates (1979) Ltd. installed kwh meters at the Betty Brite plant to monitor the energy consumption of different plant operations, such as washing, dry cleaning and pressing operations as well as building lighting and heating requirements. Betty Brite was responsible for recording daily and monthly energy consumption. Data obtained from these recordings is presented in Appendix H and also available on IBM-compatible 5.25 inch floppy diskettes.

7.0 DATA SUMMARY

7.1 DATA ANALYSIS

A sample of the data acquired from the data acquisition system is presented in Appendix G. The tables identify the date and time of the particular measurement and include flow rates through the evaporator and condensor, fresh water intake flow rates, hot water flow rates, all appropriate and required temperatures and an identifier (voltage) for the operation of washing machines, pumps and heat pump.

In order to complete the data base JFH has submitted all outputs received from the data acquisition system to the NRC. We observe from this particular data that the waste heat recovery system operated well during the period of the trial and did supply all hot water that was required by the plant during the demonstration phase. We observe from the binary code that the backup auxilliary hot water heating system was rarely operational in order to supply additional hot water. During the reported period of the service demonstration Betty Brite's business increased significantly over the same period last year. During this period Betty Brite succeeded in obtaining contract work for the supply of dry cleaning and laundering services to the University of Ottawa and a federal government department. The washing machine operation was continuous during the service demonstration and would normally not have been representative of the washing requirements that occur normally. However, the waste heat recovery system using only the heat pump without heat exchanger was capable of supplying all the necessary hot water requirements.

During the demonstration phase the waste heat recovery system experienced two compressor motor failures. The compressor motor was replaced by the supplier and sent to Fedders for an evaluation. The second unit failed a day later and was replaced by the supplier under warranty. Fedders

reported that the first compressor failure was a result of inconsistent power supply at the Betty Brite plant. During this period it is believed that the electrical supply from Ottawa Hydro may have dropped to the point that the electric motor driving the compressor was unable to cope with the demand associated with start-up of the compressor motor and resulted in burnt out. The replacement compressor failed within the first day and was judged to have been caused as a result of a shipping problem with the compressor. No other equipment failures have been reported by Betty Brite since replacement of the compressor.

During the data analysis, JFH observed erratic temperature readings for the supply of hot water from the storage tanks. Namely, delivery temperature would drop from the nominal 66°C to approximately 20°C even though the storage temperature remained at the nominal value. Further investigation of this phenomena revealed that city main water was being directly supplied to the washing machine on demand rather than as feedwater to the storage tanks. The storage tanks were found to have been connected in parallel rather than in series. The problem was corrected by connecting the storage tanks in series as originally intended.

Discussions held with Betty Brite representatives prior to finalization of this report have indicated that they are satisfied with the operation of the waste heat recovery system since the system supplied all of their current hot water demand without requiring the use of auxiliary heaters.

7.2 SAMPLE CALCULATION OF SYSTEM COP

System COP can be readily calculated from the data as follows:

Te; Average temperature difference across condensor: 6°C
Te; Average temperature difference across evaporator: 13°C
Rc; Average flow rate of freshwater through condensor: 37.8 l/min.
Re; Average flow rate of waste vaster through evaporator: 10.8 l/min.

$$COP = \frac{Tc}{Tc - \frac{Re}{Rc} Te} = 2.63$$

The values of Tc, Te, Rc and Re are obtained from the data once the erratic readings which resulted from improper installation of the storage tanks are removed. The average COP calcuated in this manner is representative of the daily average COP expected for the system and is used in the BECC model to confirm the relative economic advantage of the system over conventional oil-fired boilers for the supply of hot water in dry cleaning and laundering plants. The payback period for the current system with COP of 2.6 is less than 3.5 years. The extended demonstration under the auspices of Betty Brite will be used to verify the economic projections and augment the experience base with the system. The final stage in the project would involve product improvement prior to installation of the system in other plants.

7.3 ENERGY END USE

Based on the data obtained from the resident data acquisition system, electrical energy consumption from Ottawa Hydro, bi-monthly billing receipts and gas consumption from consumers gas bi-monthly billing receipts determined the following average energy end use profile for the Betty Brite Plant per business day:

Total energy consumption	:	1,100 kwhr/day
Natural gas consumption (steam)	:	87.40%
Hot water production (electrical)	:	4.40%
Dry cleaning equipment	:	0.28%
Laundering equipment	:	0.05%
Pressing equipment	:	0.75%
*Plant overhead (misc. equipment)	:	7.12%
		100.00%

* includes vent fan, steam extractor, misc. motors, boiler, pumps, lighting, heating, etc.

Hot water production accounts for about 35% of the total electrical demand for the plant.

8.0 CONCLUSIONS

The primary objectives of Phase A of the project have been met. The optimum size and performance of the heat pump installation has been determined. The equipment has been selected for the proposed application which ensured successful implementation within the regulatory requirements for Canada. The economic viability, capital costs, operating costs, benefits and payback period as perceived by the plant operator were determined and confirmed during Phase B of the service demonstration. A statement of energy savings for the plant and estimates on a nation wide basis were formulated. New operating and maintenance skills and its impact upon operational management have shown that no changes are expected within the present service sector. The waste heat recovery system is feasible from an investment standpoint demonstrating a payback period of less than 3.5 years over conventional oil-fired boilers.

Based on the potential penetration of heat pump waste heat recovery systems into the service sector it is estimated that a potential energy saving of 17% could be experienced from the total service sector. In comparing this energy saving to the estimated total industrial energy demand for the year 2000 for Canada the introduction of waste heat recovery system based on heat pumps into the dry cleaning and laundry service sector could result in a .1% total reduction in energy demand for the industrial sectors in Canada.

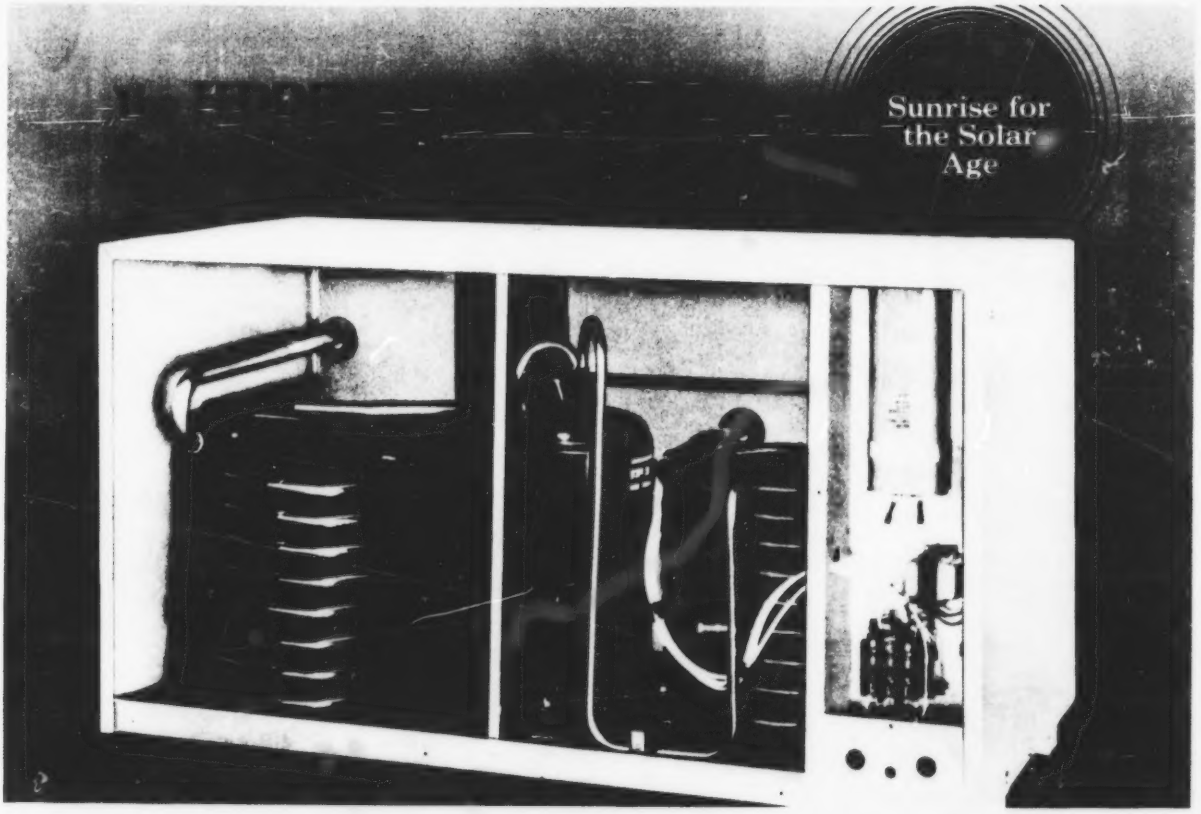
The service demonstration although experiencing some minor problems associated with the delivery of components compressor failures and installation problems has proven to be successful. The waste heat recovery system using the Fedders heat pump has been able to supply the necessary hot water requirements for typical dry cleaning and laundering plant and does so with good efficiency over conventional systems. The energy which is saved and the reduction of the operating costs associated with waste heat recovery fully substantiates the installation of such systems in other dry cleaning and laundering plants. The system has proven to be economically and technically feasible although some

modifications should be carried out to the system in order to reduce the capital cost of the unit. At the present time the capital cost of the waste heat recovery system is projected to be $8,000 (FY 82). For quantity production, the capital cost of the system should be reduced to about $6,500.00 (FY 82).

Based on the present expected rise in electrical cost being lower than those of oil, waste heat recovery will continue to be for the dry cleaning and laundering service sector an economically viable alternative. However, with the low escalation rates associated with natural gas compared to electricity waste heat recovery systems using electric heat pumps are capital intensive investments when compared to natural gas systems. In reviewing the potential market for installation of heat pumps it would appear that the likely approach which would be taken by most dry cleaning and laundering plants is conversion from oil to natural gas.

APPENDIX A

FEDDERS CARNOT II SOLAR/COMPRESSION FURNACE AND AUXILIARY COMPONENTS - TECHNICAL LITERATURE



Here is the Fedders Carnot II Solar Compression Furnace. It repositions solar radiation as the most cost-effective energy resource for many space and water heating systems...and offers a broad range of heat recovery applications.

Up to now, the wide variations in radiant insolation over the course of a year...and its unpredictable input over the course of a day imprisoned solar heating applications between an unrealistically high initial cost...and a disappointingly low solar fraction. The Solar Compression Furnace changes all that.

Connected in series to a solar collector-and-storage subsystem, Carnot II transforms solar heating into a technically practical, economically advantageous system for the first time. It raises solar storage temperatures between 60° F and 110° F to usable levels by a high-efficiency vapor compression system, rather than by supplementary resistance heat or fossil fuel as direct solar systems do.

Solar heated source water pumped through a coaxial counter-flow evaporator discharges heat to the refrigerant which is then raised to a higher temperature in the compressor. From there, it passes into a coaxial condenser from which it discharges high temperature water into a closed loop serving a terminal heating unit or other load. Similar equipment has been described as a "non-reversible heat pump". It should be noted, however, that in addition to lacking a reversing valve, there is no defrost requirement and no defrost controls, that the unit operates indoors and is largely independent of outdoor conditions, that no outdoor thermostatic control of supplementary heat is required.

A WIDE RANGE OF OPERATING CONDITIONS

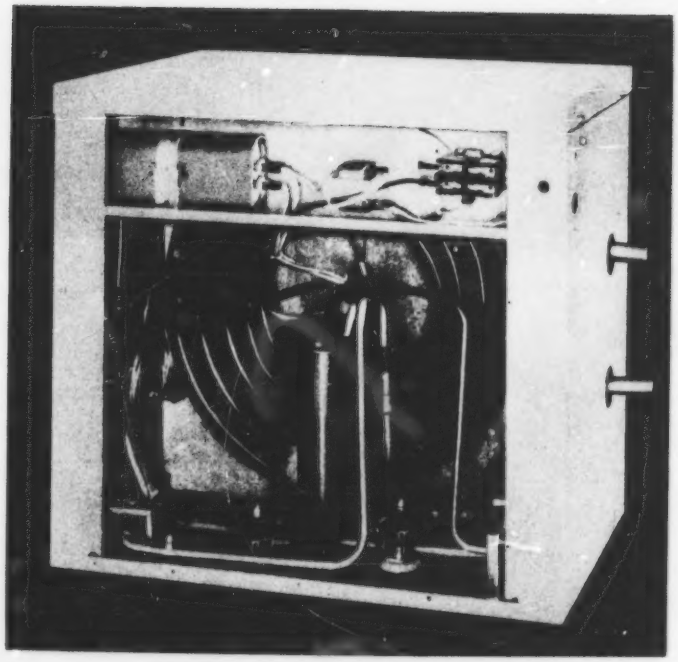
At high solar source temperatures when the Solar Compression Furnace operates at elevated evaporator temperatures, its efficiency increases. In contrast to the operation of conventional compression cycles, its watthour input actually declines. This reflects unique performance characteristics of the Carnot II's rotary compressor.

At low source temperatures, when stored energy has not been replenished due to extended overcast conditions, the Solar Compression Furnace resorts to groundwater as source energy. At an entering temperature of 55°F, groundwater from wells, ground grids or city mains permits the Solar/Compression Furnace to generate hot water with a COP of 4.2.

This defines the minimum operating efficiency of the unit and permits it to dispense completely with such back-up energy sources as resistance heat and fossil fuels which rob direct solar systems of efficiency. In systems properly sized to local levels of insolation, use of groundwater should be of brief duration and impose no significant cost penalty at this unusually high minimum efficiency. The absence of a regular energy source as a back-up will make the Solar Compression Furnace compatible to utility companies which display concern about the extra load that direct solar installations may impose during their peak operating conditions.

LOW WORKING TEMPERATURES

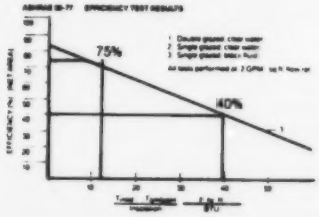
Carnot II's ability to use solar storage temperatures between 60°F and 110°F as a source without the aid of purchased thermal energy confers other important benefits on contractors and owners. Operation of solar subsystems at these low temperatures assures maximum absorption of radiant energy.



Season	Inlet Temperature 60°	75°	110°	115°	180°	215°
WINTER Average Ambient Temperature = 25° F		29.5	19.2	10.5	4.1	0.2	0.0
SUMMER Average Ambient Temperature = 75° F	X		39.4	28.1	17.5	9.3	3.1

This is apparent in standard certified ratings for solar collectors which list daily BTU output at six entering temperatures of heat transfer material as it is pumped into the collector from storage. At inlet temperatures of 75°F and 110°F, which approximate the Solar Compression Furnace's operating range, collector output is from 100% to 300% greater than at the higher collector temperatures required by "direct" solar systems.

This advantage can be seen also in collector efficiency curves which plot a ratio of the fluid-ambient T/insolation against collector efficiency on the vertical axis. The closer incoming fluid is to outdoor winter temperatures, the more solar energy is absorbed. The difference in efficiency is inversely reflected in square feet of collector surface required; the lower the efficiency, the greater the collector area. The Solar Compression Furance can therefore reduce solar collector costs to a significant degree



HEAT RECOVERY CAPABILITY

The ability of the Solar Compression Furnace to operate efficiently at high evaporating temperatures equips it for a wide variety of heat recovery applications. In commercial and industrial systems where waste process heat can supply a no-cost continuous supply of source energy, the

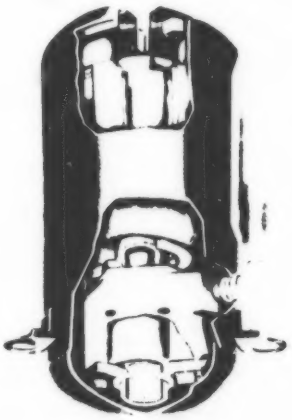
Solar Compression Furnace can make an important bottom-line contribution in business as varied as fast food restaurants and casting foundries.

If heating or cooling of process liquids is required, such liquids can be pumped directly through the evaporator or condenser which are both equipped with cupro nickel inner tubes in all models of the compression furnace. Brines, degreasing solutions and a variety of mildly corrosive process liquids can be handled directly without the losses associated with intermediate heat exchangers.

Heat recovery with the Solar Compression Furnace can be far less costly and far more flexible in many installations where air-to-air systems are the only alternative. Massive ducts, high power blowers, heat wheels and other exchangers required in air-to-air exchange can be avoided by a ¾-inch type "L" copper pipe line and a compact Solar Compression Furnace which can be installed remotely wherever needed.

THE ROTARY COMPRESSOR

The high COP's of the Solar Compression Furnace reflects performance of its rolling piston rotary compressor. This has been confirmed by independent researchers in a Federally financed DOE contract.*



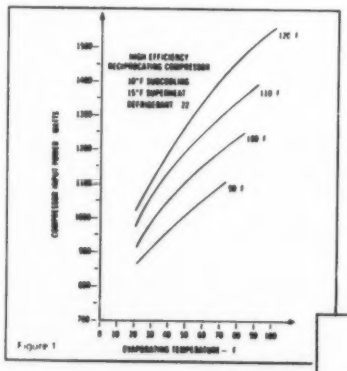


Figure 1

much as 40% higher for reciprocating compressors, accounts for Carnot II's remarkable operating efficiency.

In the graph above, typical reciprocating compressors display a sharp increase in power input for different condensing temperatures as evaporating temperatures increase. Under identical conditions, power input for the Solar Compression Furnace's rotary compressor in the lower graph declines as evaporating temperatures increase. This difference in energy consumption which can amount to as

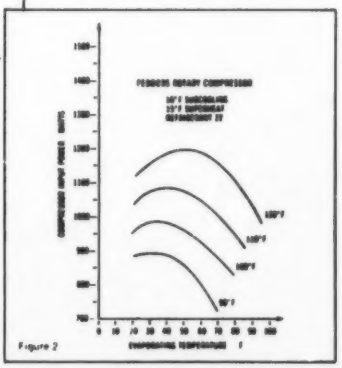


Figure 2

*Write for copy of technical paper

EQUIPMENT

The Solar Compression Carnot II Furnace is available in four capacities which are denominated in heating BTU's. They are supplied complete, ready to operate when water and power lines are connected.

They are all equipped with rotary compressors with internal overloads and counter-flow coaxial condensers and evaporators. To meet a range of operating conditions, the double wall inner tube is fabricated of corrosion resistant cupro-nickel. Automatic thermal expansion valves are factory installed. Contactors, relays and other components are factory-wired into the control compartment. The entire assembly rests on a heavy gauge steel base with structural support members and is housed in a baked enamel steel cabinet.

UNIT RATINGS

The ratings in the following tables refer only to the unit output of hot water at the indicated entering evaporator water. They do not include the significant solar fraction when collector temperatures exceed the required condensing temperature and solar radiated water moves directly to the condenser bypassing the vapor compression cycle.

STEADY STATE CAPACITY, COP EQUIPMENT RATINGS

MODEL SOCF020-700

LEAVING WATER TEMP. °F								ENTERING WATER TEMP.
90°	100°	110°	120°	130°	140°	150°	160°	
MBH	MBH	MBH	MBH	MBH	MBH	MBH	MBH	
COP	COP	COP	COP	COP	COP	COP	COP	
FPS=4; GPM=4.4; ΔPSI=1.4 ; FT. HEAD = 3.23								
143	134	126	119	108	103	96	92	55°
4.52	3.84	3.3	2.89	2.45	2.2	1.96	1.8	
176	167	158	144	138	132	124	119	70°
5.31	4.52	3.89	3.24	2.87	2.56	2.23	2.03	
198	189	180	165	159	148	143	138	80°
5.85	4.98	4.28	3.58	3.16	2.71	2.43	2.2	
220	211	203	188	181	169	164	159	90°
6.41	5.45	4.7	3.95	3.47	2.96	2.65	2.39	
	234	227	211	205	191	185	181	100°
	5.98	5.16	4.35	3.81	3.26	2.89	2.61	
		268	236	230	214	210	204	110°
		6.34	4.8	4.2	3.57	3.18	2.84	
			264	260	239	234	229	120°
			5.3	4.71	3.92	3.48	3.12	
FPS=6; GPM=6.5; ΔPSI=2.7 ; FT. HEAD = 6.24								
147	137	129	122	110	105	98	95	55°
4.74	3.98	3.4	2.95	2.48	2.22	1.97	1.8	
183	173	164	158	142	136	127	122	70°
5.75	4.81	4.09	3.58	2.95	2.61	2.27	2.05	
207	197	188	172	165	159	148	143	80°
6.45	5.39	4.58	3.77	3.3	2.91	2.49	2.24	
232	223	214	197	190	187	170	165	90°
7.29	6.05	5.12	4.24	3.69	3.31	2.75	2.46	
	250	241	224	217	201	195	189	100°
	6.78	5.73	4.78	4.13	3.48	3.06	2.72	
		271	254	247	227	220	214	110°
		6.46	5.4	4.64	3.88	3.39	3.01	
			285	278	255	249	243	120°
			6.00	5.21	4.33	3.8	3.35	
FPS=8; GPM=8.7; ΔPSI=4.3 ; FT. HEAD = 9.93								
149	139	131	123	112	106	99	95	55°
4.89	4.08	3.46	2.99	2.5	2.24	1.98	1.81	
187	177	167	159	145	139	129	124	70°
6.05	5.0	4.21	3.61	3.0	2.65	2.29	2.07	
213	203	193	176	169	163	150	145	80°
6.92	5.7	4.78	3.91	3.38	2.97	2.53	2.27	
240	230	220	203	196	189	174	168	90°
7.89	6.47	5.41	4.45	3.84	3.35	2.81	2.51	
	260	251	232	224	206	200	193	100°
	7.46	6.18	5.06	4.33	3.6	3.15	2.79	
		284	264	257	234	227	221	110
		7.08	5.82	4.96	4.07	3.54	3.12	
			299	292	265	258	252	120
			6.67	5.65	4.64	4	3.51	

MODEL SOCF040-700-800

LEAVING WATER TEMP. °F								ENTERING WATER TEMP.
90°	100°	110°	120°	130°	140°	150°	160°	
MBH	MBH	MBH	MBH	MBH	MBH	MBH	MBH	
COP	COP	COP	COP	COP	COP	COP	COP	
FPS=4; GPM=8.7; ΔPSI=1.3 ; FT. HEAD = 3.00								
319	315	309	303	296	287	277	265	55°
4.42	3.95	3.57	3.28	3.02	2.8	2.61	2.43	
372	370	366	362	355	349	340	331	70°
4.97	4.39	3.93	3.58	3.28	3.02	2.8	2.61	
410	408	404	402	398	391	383	375	80°
5.41	4.72	4.21	3.81	3.48	3.19	2.94	2.73	
448	447	446	443	440	435	430	422	90°
5.91	5.12	4.53	4.06	3.69	3.38	3.11	2.88	
	490	489	488	487	483	475	472	100°
	5.6	4.89	4.37	3.96	3.6	3.29	3.04	
		535	537	535	530	527	524	110°
		5.33	4.74	4.25	3.84	3.5	3.23	
			585	586	581	579	573	120°
			5.12	4.58	4.11	3.75	3.42	
FPS=6; GPM=13.1; ΔPSI=2.6 ; FT. HEAD = 6.01								
335	331	324	318	311	302	292	279	55°
4.71	4.17	3.74	3.4	3.13	2.89	2.67	2.49	
394	391	388	381	376	368	359	349	70°
5.43	4.72	4.18	3.76	3.43	3.14	2.9	2.68	
434	433	429	426	422	415	406	397	80°
6.02	5.17	4.53	4.05	3.67	3.34	3.06	2.83	
478	477	475	473	470	465	457	449	90°
6.73	5.7	4.96	4.4	3.95	3.58	3.27	3	
	526	523	523	522	517	511	504	100°
	6.38	5.43	4.79	4.28	3.86	3.5	3.21	
		576	576	575	573	567	561	110°
		6.05	5.27	4.65	4.18	3.78	3.44	
			630	630	628	623	619	120°
			5.8	5.1	4.53	4.07	3.69	
FPS=8; GPM=17.5; ΔPSI=4.2 ; FT. HEAD = 9.70								
343	339	333	327	320	310	298	285	55°
4.87	4.29	3.83	3.47	3.18	2.93	2.71	2.51	
404	401	397	393	387	380	370	359	70°
5.68	4.9	4.32	3.88	3.51	3.21	2.95	2.72	
448	446	442	439	435	428	420	411	80°
6.38	5.41	4.71	4.19	3.78	3.43	3.14	2.89	
496	494	491	488	485	480	473	465	90°
7.29	6.07	5.2	4.57	4.09	3.69	3.36	3.08	
	546	545	543	540	534	528	521	100°
	6.89	5.81	5.04	4.47	4	3.61	3.3	
		602	601	596	593	587	581	110
		6.57	5.62	4.89	4.35	3.91	3.55	
			659	657	651	646	642	120
			6.29	5.44	4.76	4.24	3.83	

* Instantaneous output for Solar Compression Furnace only. Does not include solar fraction.

STEADY STATE CAPACITY, COP EQUIPMENT RATINGS*

MODEL SOCF060-700/800

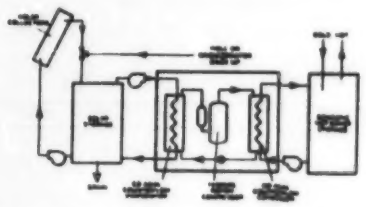
ENTERING WATER TEMP.	LEAVING WATER TEMP. °F							
	90	100	110	120	130	140	150	160
	MBH	MBH	MBH	MBH	MBH	MBH	MBH	MBH
	COP	COP	COP	COP	COP	COP	COP	COP
FPS: 4; GPM: 12.7; ΔPSI: .50 ; FT. HEAD : 1.36								
55°	443	436	429	421	411	399	384	368
	445	397	359	329	303	281	261	244
70°	517	514	508	502	494	484	473	459
	5	442	395	359	329	304	281	261
80°	569	566	562	559	552	543	533	522
	545	476	424	383	349	32	295	274
90°	623	623	619	616	611	605	597	585
	598	518	455	409	371	34	312	288
100°		683	679	681	677	670	662	656
		567	494	442	398	361	331	306
110°			747	745	742	740	733	726
			541	476	427	387	353	324
120°				818	813	812	804	798
				521	46	417	376	345
FPS: 6; GPM: 19.0; ΔPSI: 1.77 ; FT. HEAD : 4.09								
55°	464	458	450	442	432	419	404	387
	473	418	375	341	314	289	268	249
70°	546	542	535	529	521	511	499	485
	547	475	419	378	344	315	291	269
80°	604	601	596	592	584	575	564	552
	608	522	456	407	368	335	308	284
90°	663	662	660	657	653	645	634	623
	681	576	5	443	397	36	328	301
100°		730	728	728	725	718	710	699
		643	551	485	431	388	352	322
110°			801	805	801	796	787	778
			615	535	471	421	379	345
120°				878	876	873	867	859
				59	515	458	41	371
FPS: 8; GPM: 25.4; ΔPSI: 3.54 ; FT. HEAD : 8.18								
55°	476	470	462	453	443	430	414	396
	49	431	385	348	319	293	271	251
70°	561	557	551	545	537	527	513	497
	572	493	434	389	353	322	295	273
80°	621	618	613	609	603	594	583	570
	644	545	474	422	38	345	315	29
90°	688	686	680	678	673	666	657	645
	738	613	523	46	411	371	338	309
100°		759	755	752	749	742	733	721
		698	586	508	449	402	364	33
110°			835	833	829	823	815	807
			664	566	494	439	394	357
120°				915	910	905	898	893
				637	546	481	428	386

MODEL SOCF060-700/800

ENTERING WATER TEMP.	LEAVING WATER TEMP. °F							
	90	100	110	120	130	140	150	160
	MBH	MBH	MBH	MBH	MBH	MBH	MBH	MBH
	COP	COP	COP	COP	COP	COP	COP	COP
FPS: 4; GPM: 17.5; ΔPSI: 1.14 ; FT. HEAD : 2.63								
55°	545	538	528	518	506	492	475	455
	465	413	371	338	311	287	266	248
70°	639	635	628	621	612	599	584	568
	534	466	414	373	34	312	288	267
80°	704	702	696	692	685	675	661	646
	589	508	447	401	363	332	304	281
90°	777	775	769	767	761	754	744	730
	66	559	486	433	39	354	324	298
100°		850	850	849	845	845	827	818
		621	536	472	421	379	346	318
110°			933	944	941	927	917	909
			594	516	458	411	372	34
120°				1022	1021	1016	1006	999
				57	5	444	399	363
FPS: 6; GPM: 26.2; ΔPSI: 2.28 ; FT. HEAD : 5.27								
55°	569	563	553	543	530	515	496	474
	496	435	388	351	32	295	272	252
70°	671	667	661	653	644	631	615	596
	582	5	44	393	356	324	297	274
80°	744	741	735	730	723	713	699	683
	657	555	481	426	383	348	317	291
90°	824	822	815	813	807	799	788	774
	758	625	532	467	416	375	341	312
100°		910	905	902	898	890	880	865
		715	597	516	455	407	367	333
110°			1001	999	994	987	978	968
			68	577	502	445	399	361
120°				1097	1091	1086	1077	1070
				653	556	488	434	391
FPS: 8; GPM: 35; ΔPSI: 3.61 ; FT. HEAD : 8.34								
55°	583	576	566	554	542	526	507	485
	513	447	396	357	325	299	275	255
70°	690	686	678	669	659	646	629	610
	613	522	454	403	363	33	302	278
80°	767	764	758	752	744	732	717	700
	703	586	502	442	395	356	324	296
90°	849	848	844	841	835	826	811	796
	819	667	563	489	433	388	35	318
100°		941	936	935	930	918	917	898
		774	639	547	477	422	381	345
110°			1036	1036	1032	1025	1017	999
			738	618	533	468	414	373
120°				1139	1137	1132	1119	1108
				708	6	52	456	406

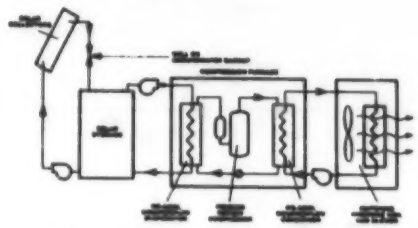
Instantaneous output for Solar Compression Furnace only. Does not include solar fraction.

SOLAR COMPRESSION WATER HEATER



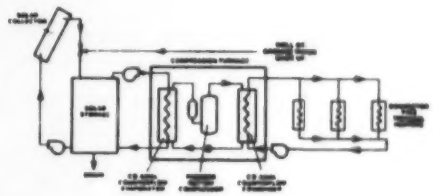
Domestic water heater utilizes solar furnace in place of heat exchanger and purchased energy to raise solar storage temperatures of 80° - 110° F to usable levels. Uses groundwater as source energy when storage temperature falls below 55° F.

SERIES-CONNECTED SOLAR COMPRESSION FURNACE



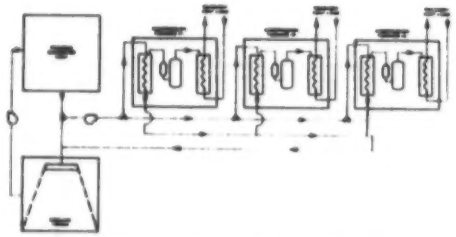
Residential heating system with fan coil in central duct plenum requires water of 120° F from solar furnace fed with 80° - 110° F storage water. Proper sizing of collector array and storage tanks minimizes reliance on 55° F groundwater.

SPACE HEATING WITH CONVECTION HEATING UNITS



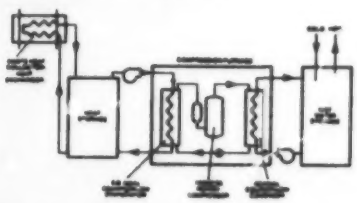
Convection-type terminal equipment (baseboard radiators, wall convectors, etc.) requiring 160° leaving water from solar furnaces, is practical retrofit for hot water central heating systems in older homes and commercial structures.

LOW-COST HEAT RECOVERY FROM COOLING TOWER



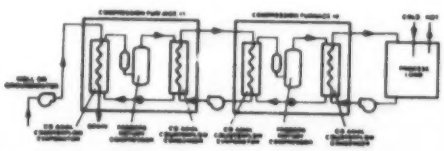
Cooling tower supply line serves as energy source for diverse remote loads without use of large costly ducts, blowers, heat exchangers, etc., required in air-to-air heat recovery systems.

WASTE HEAT RECOVERY WITH COMPRESSION FURNACE



Recapturing waste heat from exhaust hood by means of cross-flow air-to-water heat exchanger is a simple method of supplying low cost hot water for restaurants and other commercial establishments by vapor compression methods.

TWO COMPRESSION FURNACES IN SERIES UTILIZING GROUND WATER



Two series-connected compression furnaces utilize groundwater as source for process heat temperatures without solar components. COP for system is higher than that of either compression furnace, although leaving water temperatures are higher, too.

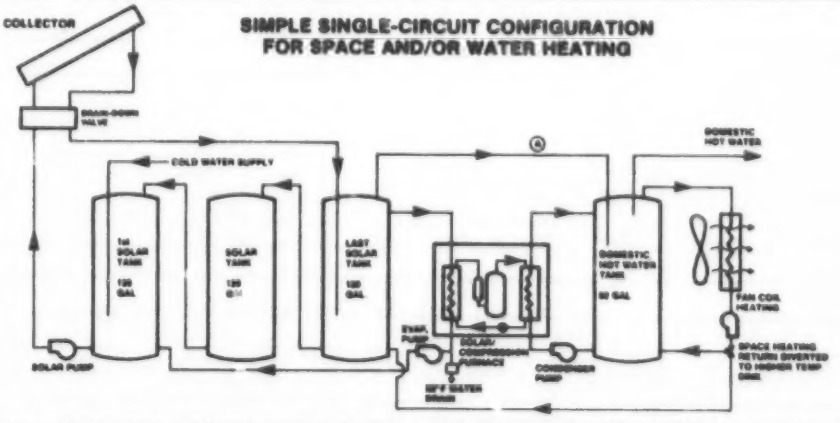
HOT WATER GENERATING CAPACITY

WATER AND/OR SPACE HEATING TEMPERATURES

Model No. Capacity @ Min. GPM	Entering Water Temp.	120°F			130°F			140°F			150°F			160°F		
		GPH	MBH	COP	GPH	MBH	COP	GPH	MBH	COP	GPH	MBH	COP	GPH	MBH	COP
	55°F	26	14.0	4.2	21	13.0	3.6	18	12.6	3.2	14	11.0	2.7	10	8.7	2.2
SOCF020	70	32	17.2	5.0	27	16.8	4.3	22	15.5	3.7	17	13.4	3.2	11	9.6	2.9
20,000 BTU	90	40	21.5	6.0	34	21.1	5.3	27	19.0	4.5	20	15.7	3.9	13	11.3	3.2
@ 2 FPS	100	45	24.2	6.7	38	23.6	5.9	30	21.1	5.2	23	18.1	4.5	15	13.1	3.7
GPM = 2.2	110				42	26.1	6.5	33	23.2	5.8	26	20.5	5.1	17	14.8	4.3
FT HEAD = 1.043 *	120							37	26.1	6.5	29	22.8	5.7	20	17.4	4.9
	55°F	51	27.5	4.3	41	25.5	3.5	34	23.9	2.9	28	22.0	2.5	23	20.0	2.1
SOCF040	70	62	33.4	5.0	52	32.3	3.8	45	31.7	3.4	38	29.9	3.1	33	28.8	2.7
40,000 BTU	90	78	42.0	5.8	62	38.5	4.7	55	38.8	4.0	49	38.6	3.8	44	38.3	3.4
@ 2 FPS	100	90	48.5	6.8	72	44.8	5.3	59	41.6	4.3	53	41.7	4.0	47	41.0	3.7
GPM = 4.4	110				88	54.7	6.2	63	44.4	4.9	57	44.9	4.1	52	45.3	3.8
FT HEAD = 1.006*	120							79	55.7	5.7	66	52.0	4.7	55	47.9	3.9
	55°F	70	37.7	4.4	57	35.4	3.5	47	33.1	3.0	39	30.7	2.5	32	27.9	2.1
SOCF060	70	87	46.9	5.4	74	46.0	4.2	62	43.7	3.8	53	41.7	3.2	45	39.2	2.8
60,000 BTU	90	109	58.8	6.4	88	54.7	4.9	78	55.0	4.4	67	52.8	4.0	62	54.0	3.5
@ 2 FPS	100	125	67.4	7.3	101	62.8	5.6	83	58.5	4.6	75	59.1	4.2	68	59.3	3.8
GPM = 6.4	110				117	72.8	6.6	96	67.7	5.2	80	63.0	4.3	74	64.5	4.0
FT HEAD = 342*	120							111	78.3	6.1	93	73.3	4.9	79	68.9	4.1
	55°F	87	46.9	4.8	70	43.5	3.8	58	40.9	3.1	47	37.0	2.6	39	34.0	2.1
SOCF080	70	117	63.1	6.0	93	57.8	4.9	78	55.0	4.1	66	55.2	3.5	56	48.9	2.9
80,000 BTU	90	144	77.6	7.8	118	73.4	6.0	102	71.9	5.2	89	70.1	4.5	78	68.0	3.4
@ 2 FPS	100	165	89.0	9.7	135	84.0	7.1	113	79.7	5.6	100	78.8	4.9	89	77.6	4.3
GPM = 8.8	110				155	96.4	8.6	129	91.0	6.5	109	85.9	5.3	99	86.3	4.7
FT HEAD = 877*	120							149	105.1	8.0	126	99.3	6.2	109	95.0	5.1

Integrated ratings calculated per NSBIR-79-1951. *Pressure drop for both evaporator and condenser

SIMPLE SINGLE-CIRCUIT CONFIGURATION FOR SPACE AND/OR WATER HEATING



Unique "Sweetwater" residential space and water heating configuration is a single circuit pressurized by city main which draws cold make-up water directly into the solar storage tank as domestic hot water is used; this delivers coolest inlet water to solar panels for maximum collector efficiency. Water fed to domestic water tank via line A is same temperature as in solar storage tanks rather than 55° F; it is, therefore, raised to usable level for water/space heating by the solar furnace with minimum kw input.

When storage temperature exceeds required heating level, the solar furnace is bypassed completely. When storage temperature falls to 55° F water from city main serves as energy source.

Only controls required are drain-down valve and differential thermostat in solar loop; regular domestic tank thermostat to activate both solar furnace pumps; main heating thermostat to activate blower and pump of hydronic coil in air handler and regular thermostat in solar storage to control solenoid for 55° F water drain.

With only one storage tank and no fan coil, this configuration is an ideal domestic hot water system.

ENGINEERING SPECIFICATIONS
FEDDERS SOLAR/COMPRESSION FURNACES

MODEL NO.	SOCF 030700	SOCF 040700	SOCF 040000	SOCF 060700	SOCF 060000	SOCF 060700	SOCF 060000
HEATING							
Nominal BTUH	20,000	40,000		60,000		80,000	
At 90° F source water and 120° F leaving water	20,200	38,000		55,000		72,000	
COMPRESSOR							
Type	Rotary, Rolling Piston Compressor						
Displacement, In ³	1.61	3.22		4.44		5.30	
Refrigerant	R12	R12		R12		R12	
ELECTRICAL							
Volts	230/208	230/208	240/208	230/208	240/208	230/208	240/208
Phase, Hertz	1/60	1/60	3/60	1/60	3/60	1/60	3/60
Rated Load, Amps	5.7	14.5	9.0	17.4	11.0	21.0	14.0
Locked Rotor Amps	31.5	105.0	85.4	124.0	85.4	150.0	107.5
Min. Circuit Amps	9.2	23.3	13.6	27.0	14.9	34.0	23.0
Max. Fuse Size	20	40	30	50	30	60	40
Control Voltage	24	24	24	24	24	24	24
EVAPORATOR & CONDENSER							
Type	Tube within a Tube, Co-Axial, Counterflow						
Inner Tube Construction	Dual Wall, Cupro Nickel, Water Side, Copper, Refrigerant Side						
Diameter	3/4"	3/4"		7/8"		3/4"	
Outer Tube	Steel, Baked Enamel Finish						
Water Connection Sweat Fitting	3/4"	7/8"		1-1/8"		1-1/8"	
PRESSURE DROP							
@2 FPS Velocity Ft. Head	1.043	1.006		342		877	
GPM	2.2	4.4		6.4		8.8	
@ 4FPS Velocity Ft. Head	3.129	3.018		1.362		2.630	
GPM	4.4	8.8		12.6		17.6	
@ 6FPS Velocity Ft. Head	6.257	6.036		4.084		5.260	
GPM	6.5	13.0		19.0		26.0	
@ 8 FPS Velocity Ft. Head	9.907	9.556		8.167		8.329	
GPM	8.7	17.4		25.2		34.8	
DIMENSIONS							
Length	24-1/8"	42-1/4"		42-1/4"		42-1/4"	
Height	22-3/8"	23-1/2"		23-1/2"		23-1/2"	
Depth	16-3/4"	19-1/2"		19-1/2"		19-1/2"	
WEIGHT							
Shipping (lbs.)	157	287		337		387	
Net (lbs.)	135	250		300		350	

Specifications subject to change without notice.

FEDDERS
SOLAR PRODUCTS COMPANY
a division of Fedders Corporation • Edison, New Jersey 08817

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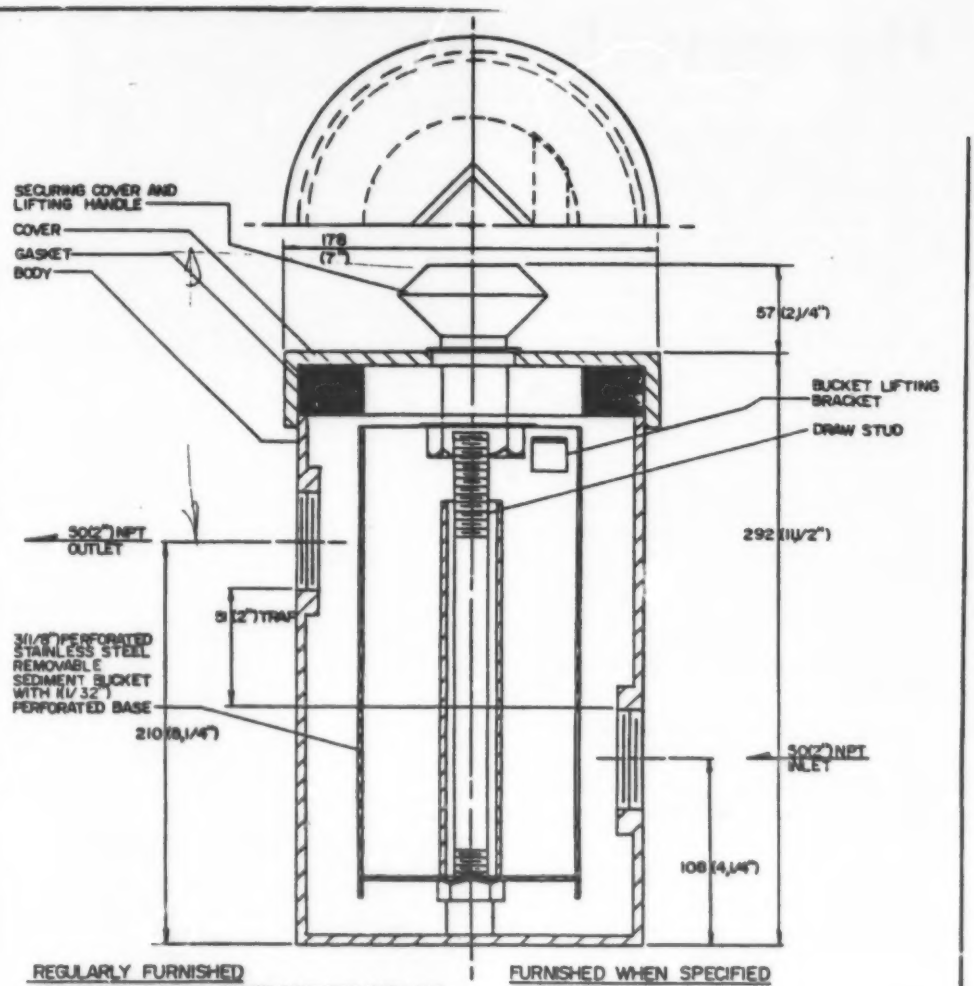


REGULARLY FURNISHED

ACID RESISTING EPOXY COATED CAST IRON BODY AND SECURED COVER, REMOVABLE STAINLESS STEEL SEDIMENT BUCKET.

FURNISHED WHEN SPECIFIED

□ - 38 (1/2") CONNECTION

FUNCTION - RECOMMENDED FOR INSTALLATION IN DRAINAGE SYSTEMS FOR INTERCEPTING HAIR, PLASTER, METALS, SEDIMENT ETC..

NOTE: ALLOW 305 (12") CLEARANCE ABOVE INTERCEPTOR FOR BUCKET REMOVAL.

DIMENSIONS ARE IN INCHES (IN BRACKETS) AND TO THE NEAREST MILLIMETRE.

JAN 16 '81 LM

HAIR AND SOLIDS INTERCEPTOR	PRODUCT NUMBER Z-1180
ZURN ZURN DRAINAGE AND CONTROL SYSTEMS LTD.	DRAWING NUMBER 0710

Honeywell

THE T675 AND T678 TEMPERATURE CONTROLLERS REGULATE THE TEMPERATURE OF AIR OR LIQUIDS IN DUCTS, PIPES, TANKS, AND BOILERS. TYPICAL USES INCLUDE CONTROL OF DAMPERS AND VALVES IN HEATING, COOLING, OR HEATING-COOLING SYSTEMS.

- T675A has 1 single-pole, double-throw (spdt) switch that breaks R-B and makes R-W at the set point on a temperature rise.

- T675B Manual Reset Low Limit has 1 single-pole, single-throw (spst) switch that breaks at the set point on a temperature fall, and requires manual resetting.

- T678A has 2 single-pole, double-throw (spdt) switches that operate in sequence. The right switch breaks R-B and makes R-W at the set point on a temperature rise. The left switch breaks R-B and makes R-W if the temperature continues to rise through the interstage differential.

- T675A models available with an adjustable differential.

- T675A and T678A have set point adjustment knob on front of case.

- Capillary tubing allows remote mounting of sensing element; models available with various lengths.

- T675A and T678A models available with a fast-response sensing element.

- Controller can be mounted in any position.

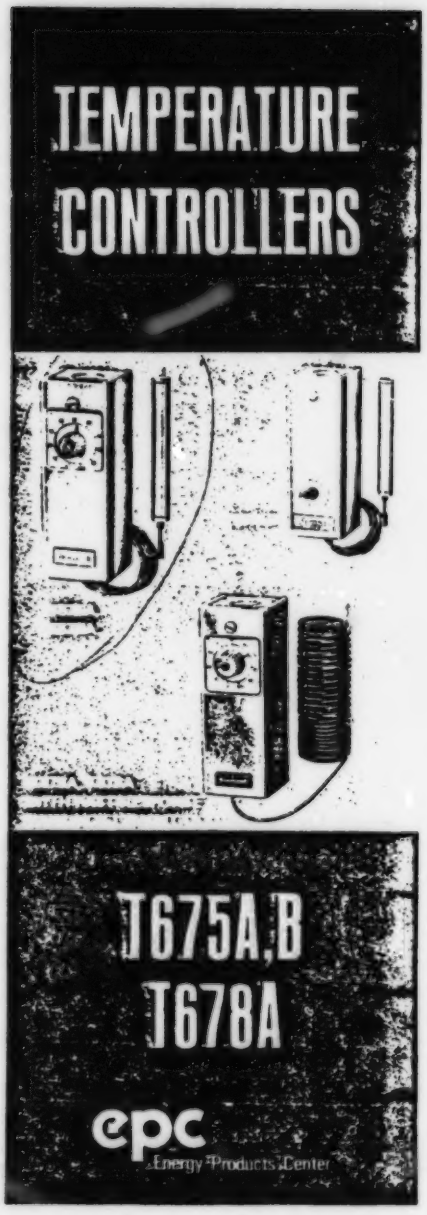
- Mounting accessories available for all applications.

- Ambient temperature compensated.

- Insert supplied with TRADELINE models replaces set point knob to discourage tampering.

G.S.
Rev. 10-81•

Form No.
60-2200—6



SPECIFICATIONS

TRADELINE MODELS

TRADELINE models are selected and packaged to provide ease of stocking, ease of handling, and maximum replacement value. TRADELINE model specifications are the same as those of standard models except as noted below.

TRADELINE MODELS AVAILABLE:

T675A and T678A Temperature Controllers—see Table I for TRADELINE models available.

CAPILLARY: 5 ft [1.5 m] long, copper (except 1 T675A model with a 20 ft [6.1 m] capillary).

TRADELINE FEATURES:

- All TRADELINE Models include tamper-resisting insert and 107324A Capillary Holder. (Models with fast-response sensing element include 131524A Capillary Holder.)
- TRADELINE models of T675A, T678A supplied with impact-resistant Norel covers.
- TRADELINE pack with cross reference and special instruction sheet.

TABLE I—TRADELINE MODELS AVAILABLE

MODEL	DIFFERENTIAL	OPERATING TEMPERATURE RANGE (SCALE RANGE)		MAXIMUM SAFE BULB TEMPERATURE	
		F	C	F	C
T675A (1 spdt switch)	Adjustable 3 F to 10 F [1.7 C to 5.6 C]	0 to 100[a,b]	-18 to + 38	125	52
		160 to 260	71 to 127	280	138
	Adjustable 3.6 F to 12 F [2.0 C to 6.7 C]	55 to 175[a]	13 to 79	200	93
	Fixed at 1 F [0.6 C]	0 to 100	-18 to + 38	125	52
T678A (2 spdt switches)	Each switch: Fixed at 3 F [1.7 C] Interstage: Adj. 3 F to 10 F [1.7 C to 5.6 C]	0 to 100[a]	-18 to + 38	125	52
	Each switch: Fixed at 3.6 F [2.0 C] Interstage: Adj. 3.6 F to 12 F [2.0 C to 6.7 C]	55 to 175[a]	13 to 79	200	93

[a]TRADELINE model also available with a fast-response sensing element, including a 131524A Capillary Holder.
[b]TRADELINE model also available with a 20 ft [6.1 m] copper capillary.

ORDERING INFORMATION

WHEN PURCHASING REPLACEMENT AND MODERNIZATION PRODUCTS FROM YOUR TRADELINE WHOLESALER OR YOUR DISTRIBUTOR, REFER TO THE TRADELINE CATALOG OR PRICE SHEETS FOR COMPLETE ORDERING NUMBER, OR SPECIFY—

1. Order number; specify TRADELINE model, if desired.
2. Operating temperature range (scale range).
3. Standard sensing bulb or fast-response sensing element (if available).
4. Capillary length and material.
5. Fixed or adjustable differential on a T675A.
6. Differential range on adjustable T675A and T678A models.
7. Replacement parts, if desired.
8. Accessories, if desired.

IF YOU HAVE ADDITIONAL QUESTIONS, NEED FURTHER INFORMATION, OR WOULD LIKE TO COMMENT ON OUR PRODUCTS OR SERVICES, PLEASE WRITE OR PHONE:

1. YOUR LOCAL HONEYWELL RESIDENTIAL SALES OFFICE (CHECK WHITE PAGES OF YOUR PHONE DIRECTORY).
2. RESIDENTIAL GROUP CUSTOMER SERVICE
   HONEYWELL INC., 1885 DOUGLAS DRIVE NORTH
   MINNEAPOLIS, MINNESOTA 55422 (612) 542-7500

(IN CANADA—HONEYWELL CONTROLS LIMITED, 740 ELLESMERE ROAD, SCARBOROUGH, ONTARIO) INTERNATIONAL SALES AND SERVICE OFFICES IN ALL PRINCIPAL CITIES OF THE WORLD.

STANDARD MODELS

MODELS (also refer to Table II):

T675A Temperature Controller–spdt switching to make or break a circuit on a temperature change; fast-response models operate approximately 4 times faster than standard models.

T675B Low Limit Temperature Controller–spst switching to break a circuit on a temperature fall; must be manually reset.

T678A Temperature Controller–2 spdt switches operate 2 independent circuits in sequence; fast-response models operate approximately 4 times faster than standard models.

TABLE II–STANDARD MODELS AVAILABLE

MODEL	SNAP-ACTING SWITCHES	DIFFERENTIAL[a]	OPERATING TEMPERATURE RANGE (SCALE RANGE)[c]		MAXIMUM SAFE BULB TEMPERATURE	
			F	C	F	C
T675A	one spdt	Adjustable 3 F to 10 F [1.7 to 5.6 C]	0 to 100	-18 to + 38	125	52
			160 to 260	71 to 127	280	138
		Adjustable 3.6 F to 12 F [2.0 to 6.7 C]	55 to 175	13 to 79	200	93
		Fixed at 1 F [0.6 C]	0 to 100	-18 to + 38	125	52
			55 to 175[d]	13 to 79	200	93
			160 to 260	71 to 127	280	138
T675B	one spst	Manual reset[b]	30 to 50[e]	- 1 to + 10	125	52
T678A	two spdt	Each switch: Fixed at 3 F [1.7 C] Interstage: Adjustable 3 F to 10 F [1.7 C to 5.6 C]	0 to 100	-18 to + 38	125	52
			160 to 260	71 to 127	280	138
		Each switch: Fixed at 3.6 F [2.0 C] Interstage: Adjustable 3.6 F to 12 F [2.0 C to 6.7 C]	0 to 100[f]	-18 to + 38	125	52
			55 to 175	13 to 79	200	93

[a] All differentials are subtractive except for the T675B (which has manual reset) and the interstage differential on T678A models.

[b] Temperature should rise 10 F [5.6 C] above the set point before resetting.

[c] Celsius scaleplates available. See Accessories.

[d] Model available with a fast-response sensing element, including a 131524A Capillary Holder.

[e] Factory-set and locked at 37 F [3 C].

[f] Sensing element is a 12 ft [3.7 m] averaging element on the end of a 10 ft [3.0 m] copper capillary.

ELECTRICAL RATINGS:

T675A models with adjustable differential and T678A:

	120 Vac	240 Vac	277 Vac
Full Load	8.0	5.1	4.2
Locked Rotor	48.0	30.6	25.2
Pilot Duty	125 VA	125 VA	125 VA

T675A models with fixed differential–125 VA at 120/208/240/277 Vac.

T675B–125 VA pilot duty up to 277 Vac.

T678A–maximum connected load = 2000 VA.

OPERATING TEMPERATURE RANGE (SCALE RANGE): See Table II.

DIFFERENTIAL: See Table II.

ADJUSTMENTS:

Set Point (Operating Temperature):

T675A and T678A–adjustment knob on dial on on front of controller.

T675B–screwdriver slot in center of dial (cover must be removed).

Differential (cover must be removed):

T675A models with adjustable differential–adjustment dial on snap-acting switch.

T678A interstage differential–star wheel on left side of chassis.

(continued on page 4)

GRUNDFOS

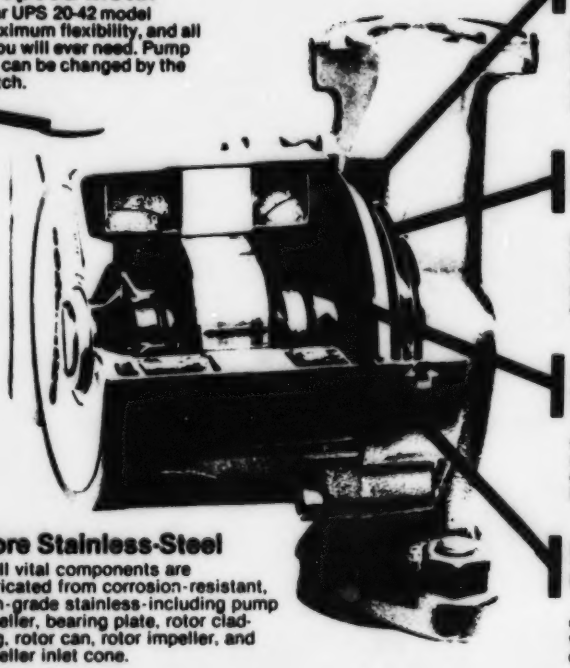
Maintenance-Free Domestic Circulators

For Closed System Heating Applications:

- Hydronic Heating
- Solar Thermal Heating
- Heat Recovery

Grundfos Domestic Circulators...the most pump for the money.

Now a 3-Speed Motor
Our popular UPS 20-42 model gives you maximum flexibility, and all the control you will ever need. Pump performance can be changed by the flick of a switch.



Enlarged Stainless-Steel Bearing-Plate
Prevents a direct interchange of fluid between pump and motor chambers. The bearing-plate's larger surface area efficiently transfers motor heat directly to the pumping medium.

Direct-Drive to Stainless Impeller
Very high pump efficiencies are achieved because virtually all motor-shaft power is transferred directly to the liquid. No power is wasted overcoming frictional losses incurred with a shaft-seal and coupling assembly.

New, Patented Pump Within the Motor
Stainless-Steel pump force-circulates lubricating fluid within the motor chamber, insuring efficient dissipation of heat and constant lubrication of bearing surfaces.

More Stainless-Steel
All vital components are fabricated from corrosion-resistant, high-grade stainless-including pump impeller, bearing plate, rotor cladding, rotor can, rotor impeller, and impeller inlet cone.

Diamond-Hard Ceramic Bearings
An ultra-hard ceramic shaft runs in ceramic radial rings—as close to a wear-free combination as modern engineering can achieve.

Over 20 years, and 20 million pumps worth of experience went into the development of these circulators...

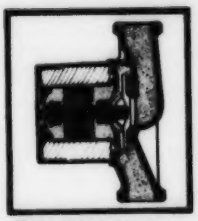
The fluid being pumped lubricates the diamond-hard pump and motor bearings.

So, there is no shaft seal to leak or wear out. No coupling assemblies to adjust or replace. No oiling—ever.

Grundfos Circulators effectively lubricate themselves.

■ **Improved lubrication system prevents system sediments from entering the rotor can.**

The water which initially fills the rotor chamber remains static and isolated from system water. There is little or no interchange of motor fluid and system fluid.



Motor bearings run cleaner, and the pump lasts longer.

■ **20% more starting torque**

Our new extra-dense rotor, and improved stator design assures you of trouble-free, high-torque start-ups every time, every season.

Each rotor is statically and dynamically balanced.

■ **Whisper-Quiet Running**

In fact, you'd have a hard time even hearing our pumps run because we have eliminated the noise causing coupling and shaft-seal assemblies.

Less than 32 dBA measured from 6"

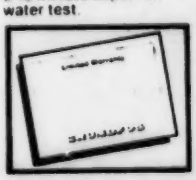
■ **Each unit individually water tested**

And tested under conditions that simulate or exceed actual working conditions. When you pull a Grundfos circulator from the box, you know it's going to work.

Every pump is put through a 15 minute super-hot water test.

■ **18 Month Warranty**

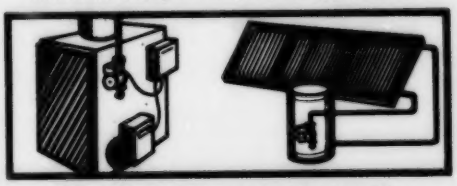
That's 18 months from the date of installation, not the date of manufacture. This is the longest pump warranty in the industry.



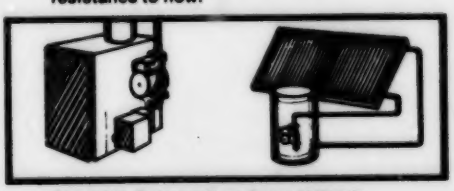
We know our pumps are built to last

The model UPS 20-42 is actually three pumps in one. Here's why this unique feature makes your job easier.

UPS 20-40 models are equipped with a three-position speed control switch. This offers you instant flexibility, and easy method for correcting velocity noise in hydronic systems. Plus, three speeds provides the close flow control necessary for solar thermal applications.



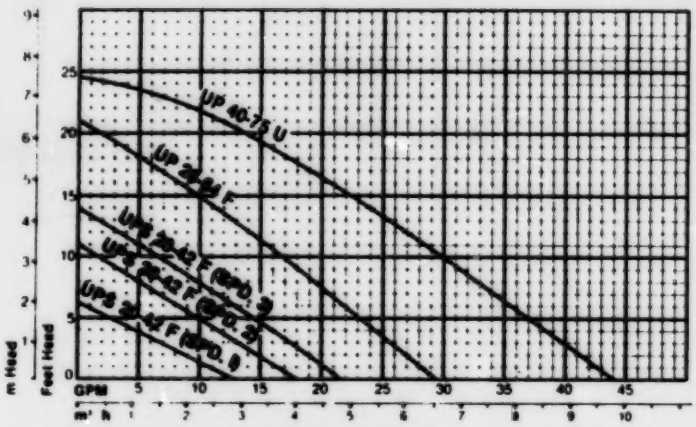
Speed III–The high speed setting should be used for light commercial or larger residential boilers, for long runs, and for larger solar applications with higher resistance to flow.



Speed II–This medium-flow setting is ideal for the majority of light-residential hydronic and solar applications.

Speed I–Low speed setting is particularly useful for multiple-zone applications where pumps are used instead of zone valves for maximum reliability and optimum flow response, and in low flow solar and heat-recovery applications.

Performance:



Applications:

Grundfos Domestic Circulators are single-stage, direct-drive, centrifugal pumps, designed primarily for closed system water applications. These pumps can be operated up to system pressures of 142 psi, with fluid temperatures of 230°F and corresponding ambient temperatures of 68°F (104°F for UP 40-75 U). 50% by volume mixtures of water and ethylene or propylene glycol solutions may be pumped. Check with Grundfos for information regarding suitability of other fluids.

Construction Materials:

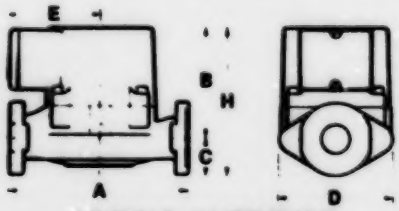
Stainless-Steel: Impeller, inlet cone, bearing plate and bearing retainers, rotor can, rotor cladding, rotor impeller, shaft retainer
Aluminum: Stator housing, terminal box.
Aluminum Oxide Ceramic: Shaft, upper and lower radial bearings.
Metal Impregnated Carbon: thrust bearing.
Cast Iron: pump housing (volute)
EP (ethylene propylene rubber): O-ring, gasket, flange gaskets.
Brass: rotor end caps

Accessories:

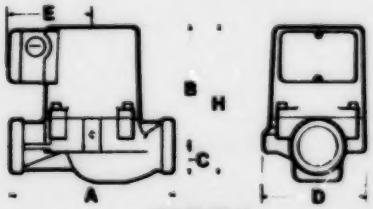
Flange Sets: ¾" to 1½"
Bronze Isolation Valves: ¾" to 1¼"
Cast Iron Half-Union Sets: 1¼"

Electrical:

Model	Speed	Hp	Watts	Volts	Amps	RPM	Capacitor
UPS 20-42 F 3-Speed	3	1/20	95	115	0.85	2620	10MF/180V
	2	1/32	70		0.60	2300	
	1	1/64	50		0.42	1800	
UP 26-64 F 1-Speed	1	1/12	185	115	1.75	3100	8MF/180V
	1	1/12	185	230	0.80	3100	2.5MF/380V
UP 40-75 U High Velocity	1	1/8	225	115	2.15	3250	10MF/180V



UP 26-64 F and UPS 20-42 F



UP 40-75 U

Dimensions and Weights:

Model	A	B	C	D	E	H	Shipping Carton LxWxH-in	Pack Vol ft³	Net Weight lbs	Ship Weight lbs
UPS 20-42 F 3-Speed	6 1/2	4 3/16	1 5/16	4 3/16	3 9/16	5 1/2	7 7/8 x 6 1/8 x 5 7/8	1/6	9	10
UP 26-64 F 1-Speed	6 1/2	5 1/16	1 5/16	4 3/16	3 9/16	6 3/8	7 7/8 x 6 1/8 x 7	1/5	12	12 1/2
UP 40-75 U High Velocity	7 1/8	5 3/16	1 9/16	4 7/16	3 5/8	6 3/4	8 3/4 x 7 3/4 x 8	1/3	12	13

GRUNDFOS

2555 Clovis Avenue • Clovis, California 93612 • Phone (209)299-9741 • Telex:35-5363

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APPENDIX B

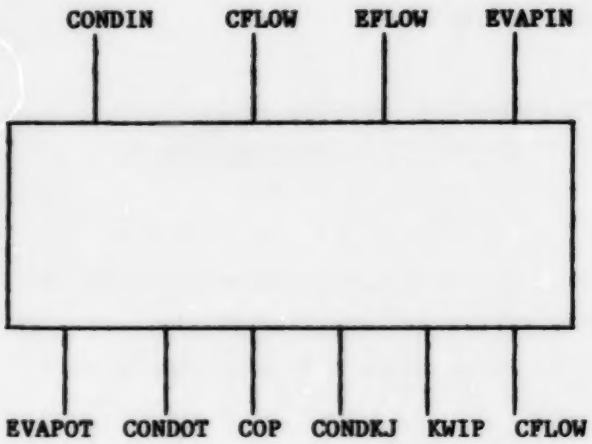
TRNSYS HEAT PUMP MODULE DESCRIPTIONS

HEAT PUMP MODULE

Performance data for the heap pump module is based on information supplied by Fedder's Corp. for their Carnot II Solar/Compression Furnace.

This information is used in three progressive linear interpolations of performance data contained in "look-up tables" supplied by the manufacturer. Therefore only outlet temperature and input energy were used.

An energy balance performed on the original data revealed that actual heat transfer was 5% lower than claimed by the tables. Because of this, the program was redesigned to calculate the heat transfer and COP based on temperature and flows from the tables.



INPUTS

CONDIN = Condensor inlet temperature, °C.

EVAPIN = Evaporator inlet temperature, °C.

CFLOW = Condensor Flow rate, kg/hr.

EFLOW = Evaporator Flow rate, kg/hr.

OUTPUTS

EVAPOT = Evaporator outlet temperature, °C.

CONDOT = Condensor outlet temperature, °C.

COP = Condensor COP.

CONDKJ = Heat transfer to condensor, kJ.

KWIP = Energy used by heat pump, kJ.

CFLOW = Condensor Flow rate, kg/hr.

DIRTY WATER STORAGE TANK MODULE

The dirty water storage tank module is a constant volume tank with an integral flow mixer and heat exchanger. The program is arranged so that any incoming flow of water is mixed with existing volume, before heat transfer calculations are made. The storage tank has been designed as a constant volume unit with the user initiating the volume at the start of the program.

The program is structured with the following order:

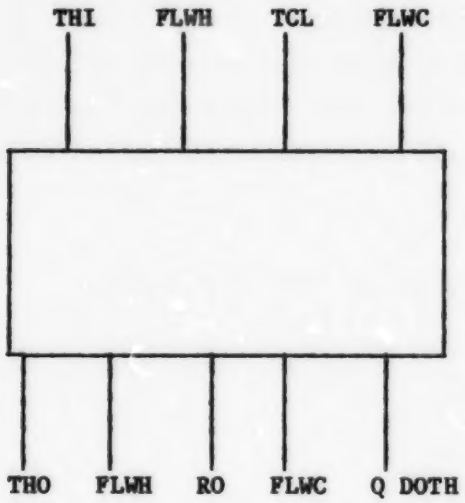
1. Mixes incoming volume and temperature with existing volume and tank temperature.

2. Calculates heat loss of the water to the tank and atmosphere.

3. When there is flow through the cold side of the heat exchanger, the program calculates the heat loss from the waste water to the feed water based on the efficiency of the heat exchanger.

4. All temperatures are then recalculated.

Depending on how short the timestep is, the more accurate the calculation.



PARAMETERS:

VCONST = Volume of "Hot" liquid in the tank, m^3.

HTC = Heat transfer coefficient for loses to the environment

TENV = Temperature of the environment, °C.

CP = Specific heat capacity of the fluids, kJ/kg°k.

DENSITY = Density of the fluids, kg/m^3.

EFFECT = Constant effectiveness of the Heat exchanger, %.

TINT = Initial temperature of hot side storage, °C.

AREA = Surface are of hot side of the tank, m^2.

INPUTS:

THI = Temperature of incoming hot liquid, °C.

FLWH = Mass flow rate of incoming hot liquid, kg/hr.

TCI = Temperature of incoming cold liquid, °C.

FLWC = Mass flow rate of incoming cold liquid, kg/hr.

OUTPUTS

THO = Average tank temperature, °C.

FLWH = Mass flow rate of leaving hot liquid, kg/hr.

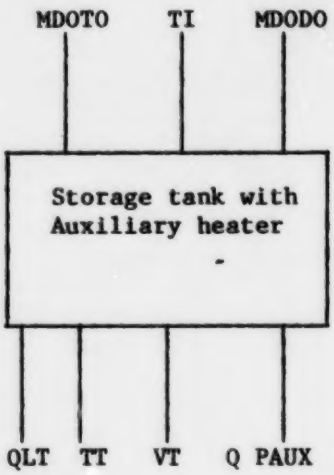
TCO = Temperature of leaving cold liquid, °C.

FLWC = Mass flow rate of cold liquid, kg/hr.

Q DOTH = Rate of energy transfer to cold liquid, kj/hr.

CLEAN WATER STORAGE TANK MODULE

The clean water storage tank module is also a constant volume tank. It has an integral heater that supplies makeup heat to the system or all of the heat if the heat pump is not functioning. The simulation works by first mixing the incoming volume with the existing volume and finding the average tank temperature. The program then calculates the heat losses to the environment of the tank then recalculates a new tank temperature. If this temperature is below the set turn on temperature of the heater, the program turns on the heater and adds heat at a user specified heat rate. It then recalculates the new tank temperature.



PARAMETERS

QCS = Heat loss coefficient through the sides of the tank, kj/hr m^2°C.

QCB = Heat loss coefficient through the bottom of the tank, kj/hr^2°C.

TENV = Temperature of the environment, °C.

TDIA = Diameter of the tank, m.

CPF = Specific heat of the fluid in the tank, kj/kg.k.

RHO = Density of fluid in the tank, kg/m^3.

VTINIT = Initial volume of fluid in the tank, m^3.

TINIT = Initial temperature of fluid in tank, °C.

AUXPWR = Power capacity of auxiliary heater is turned, on kj/hr.

AUXLOW = Temperature at which auxiliary heater is turned on, °C.

AUXHI = Temperature at which auxiliary heater is turned, off, °C.

AUXINT = Initial setting of auxiliary heater, (i10 = ON/OFF).

INPUTS

MPOTI = Mass flow rate into the tank, kg/hr.

MDOTO = Mass flow rate out of the tank, kg/hr.

TI = Temperature of water flowing into the tank, °C.

OUTPUTS

UT = Volume of water in the tank, m^3.

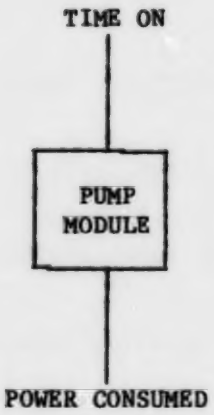
TT = Average tank temperature, °C.

QLT = Energy loss to the environment, kj/hr.

QPAUX = Energy added by auxilary heater, kj/hr.

AUXILIARY PUMP MODULE

The auxiliary pump works in conjunction with the heat pump. When the heat pump is running, a function tells the pump module to turn itself on. The module then tabulates the power consumed given the energy usage rate. The pump module works as part of the TRNSYS integrator module.



PARAMETER

Pump power consumption rate, kj/hr.

INPUT

Time in which the pump is running, hrs.

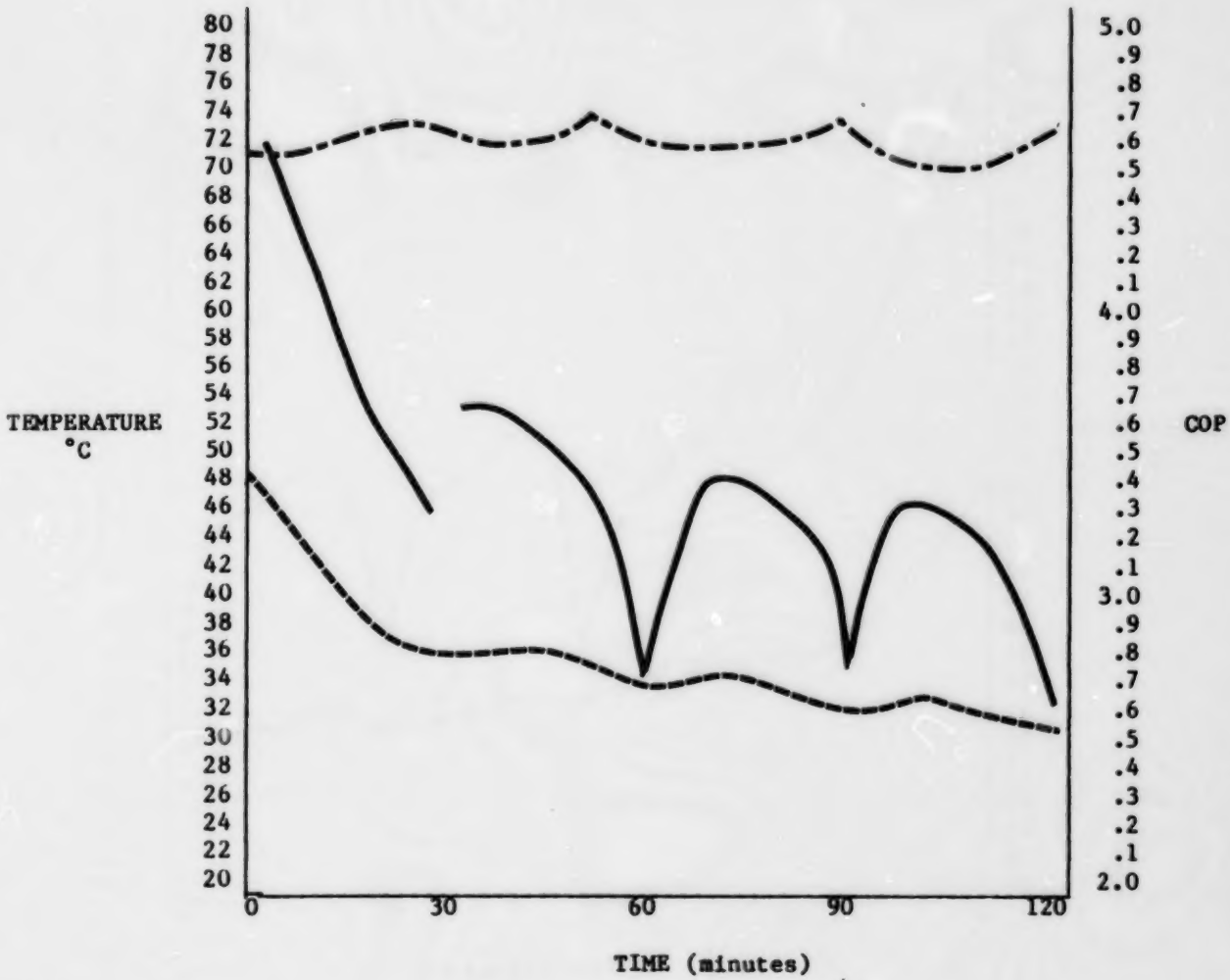
OUTPUT

Quantity of power consumed, kj.

APPENDIX C

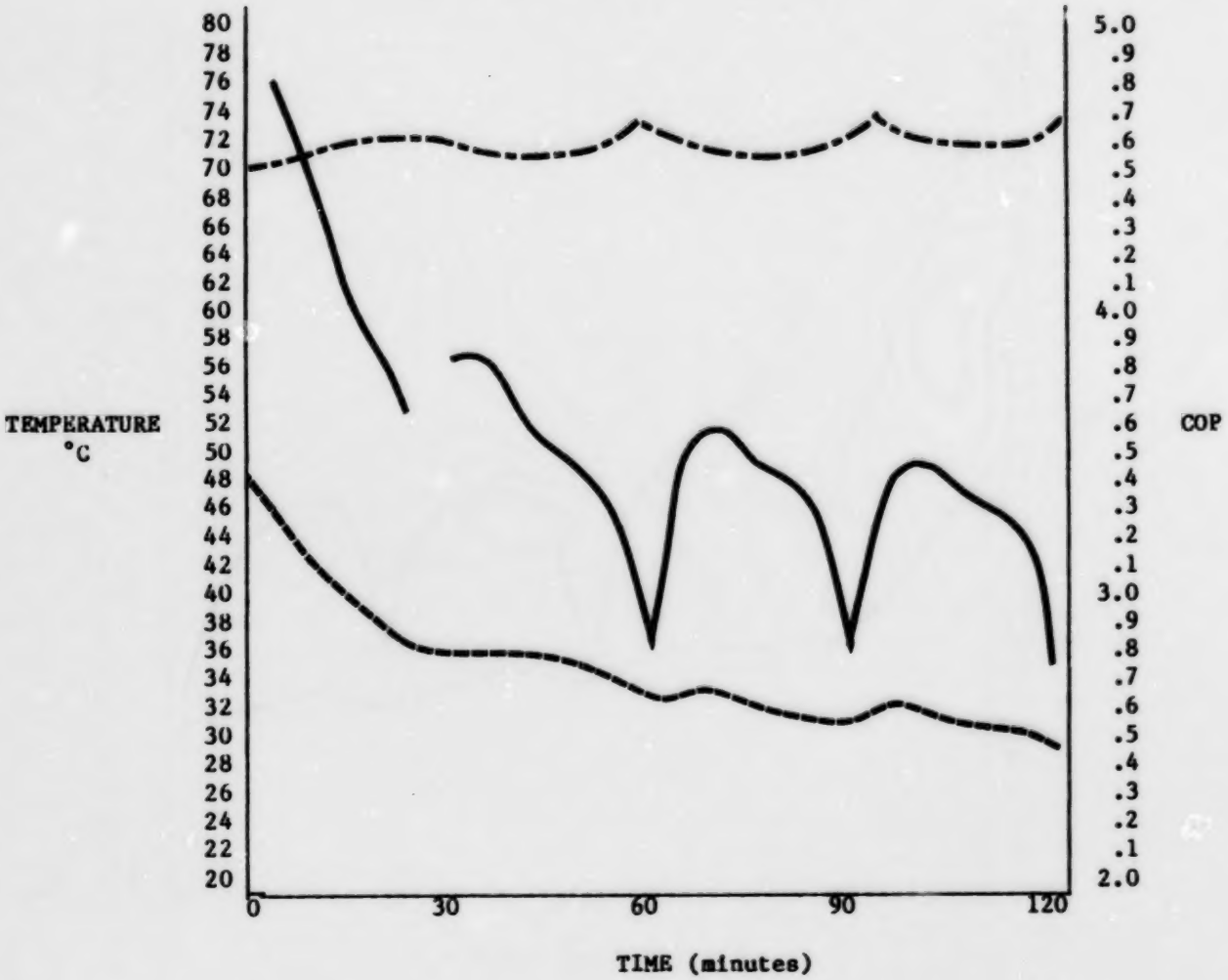
TRNSYS SIMULATION RESULTS

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



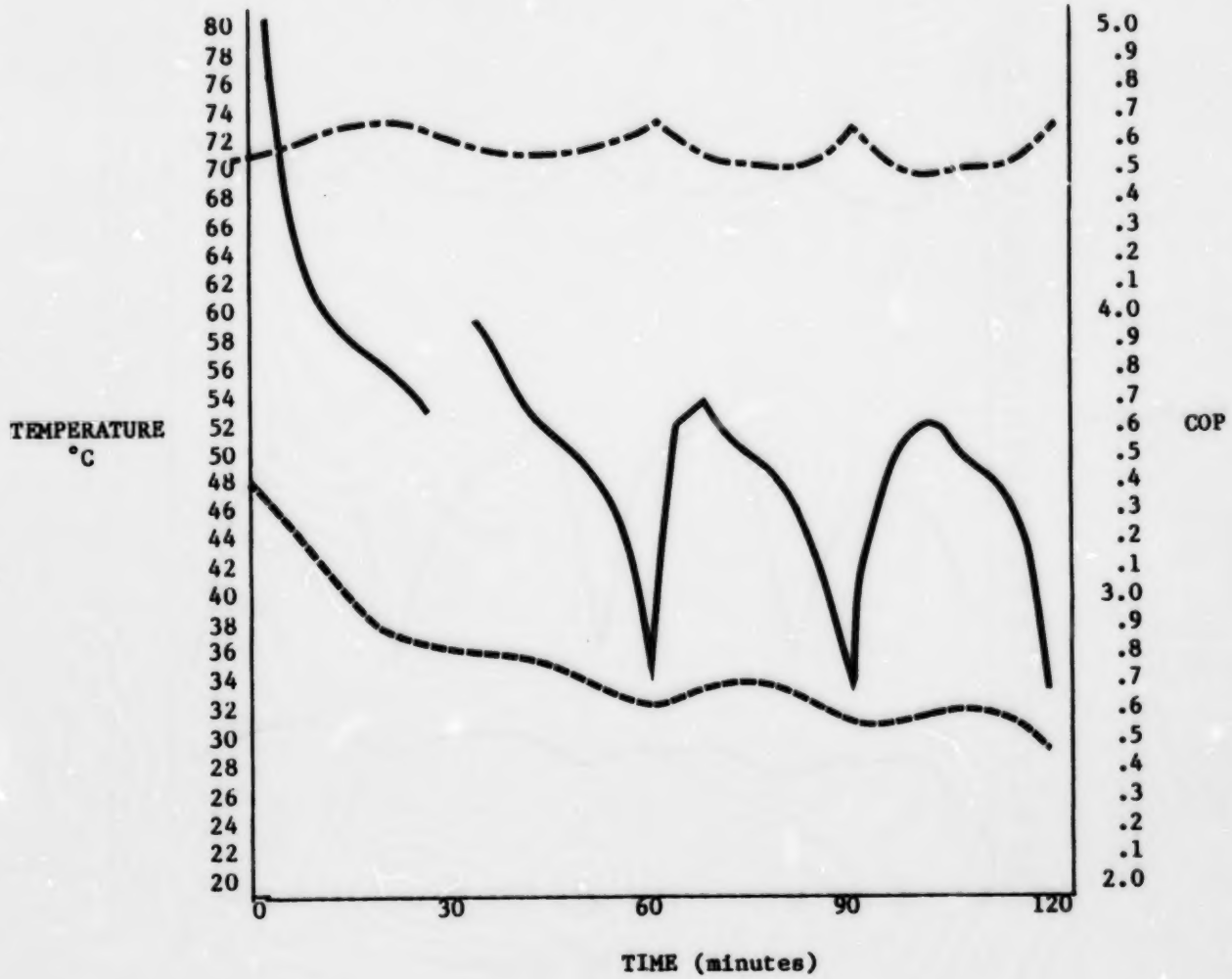
80 series Heat Pump with heat exchanger efficiency of 40%
100% utilization
TINT 48°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



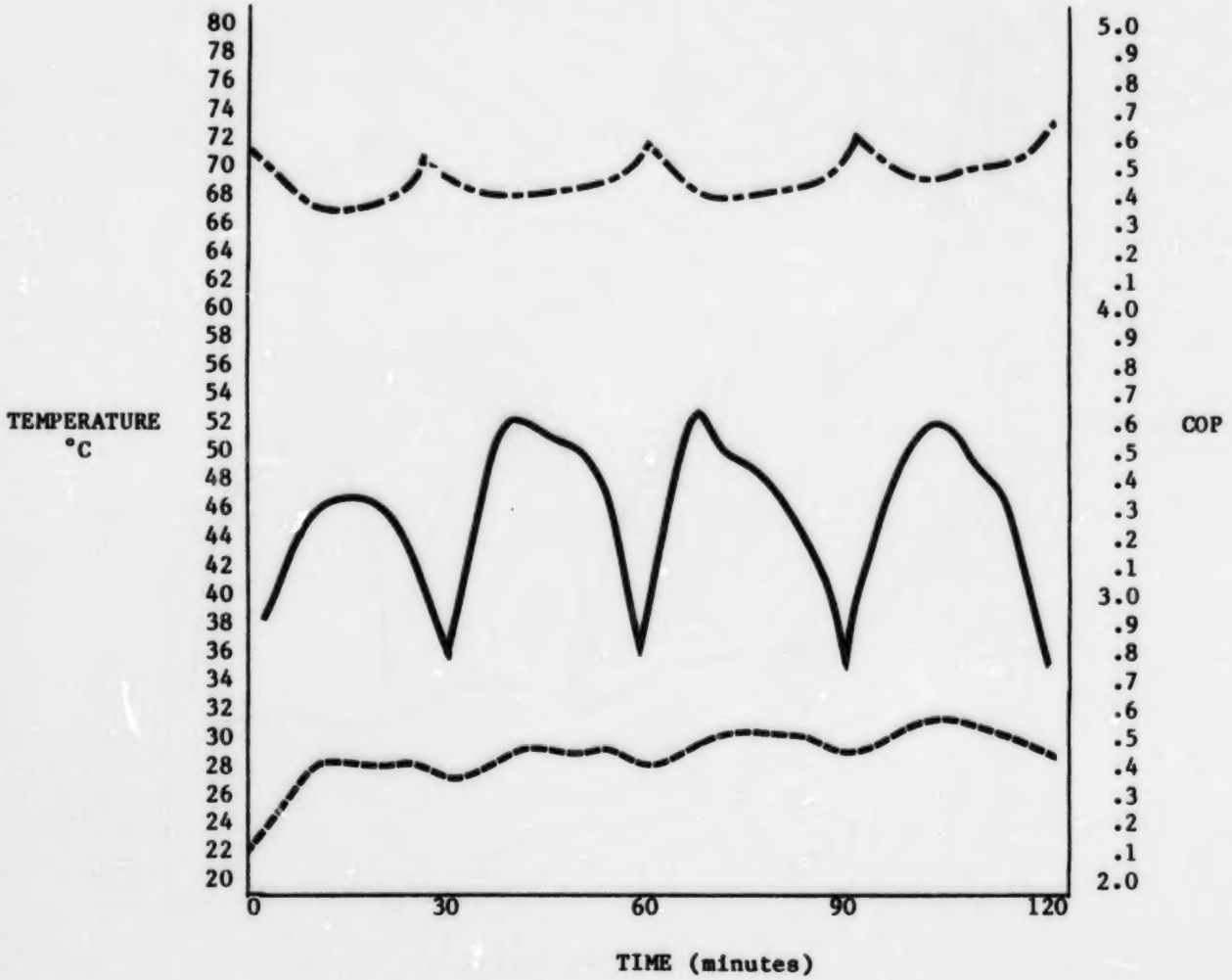
80 series Heat Pump with heat exchanger efficiency of 50%
100% utilization
TINT 48°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



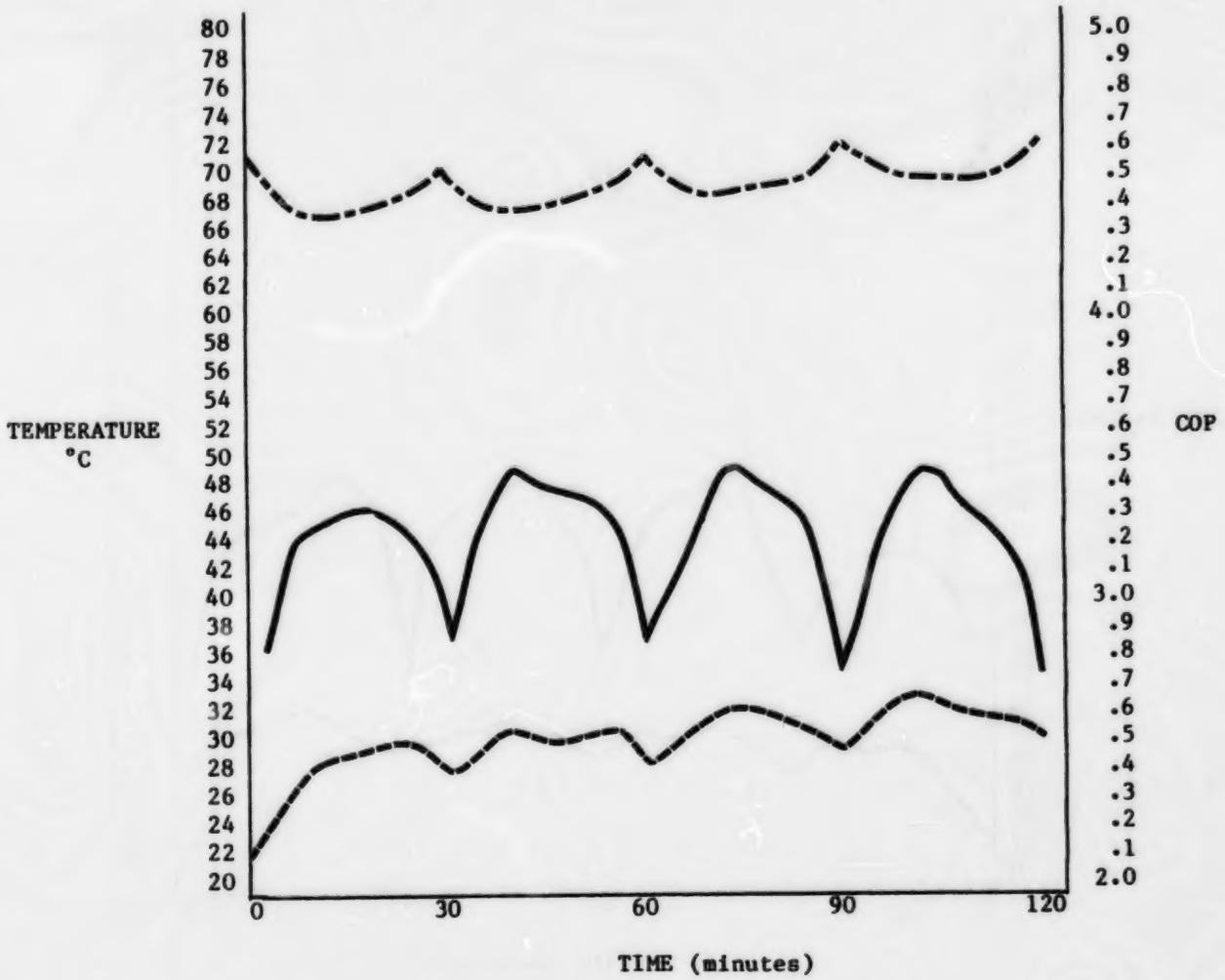
80 series Heat Pump with heat exchanger efficiency of 60%
100% utilization
TINT 48°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



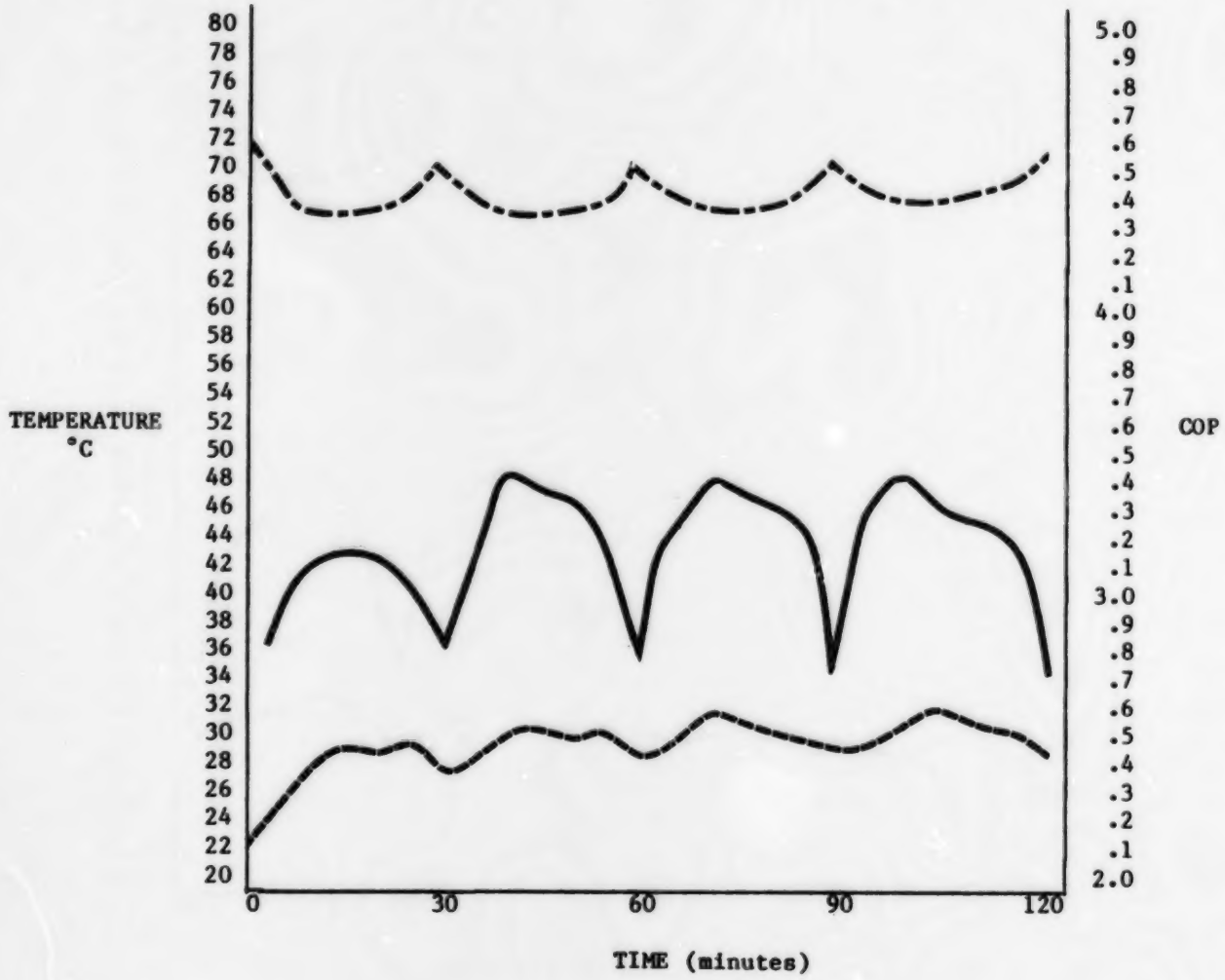
80 series Heat Pump with heat exchanger efficiency of 60%
100% utilization
TINT 22°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



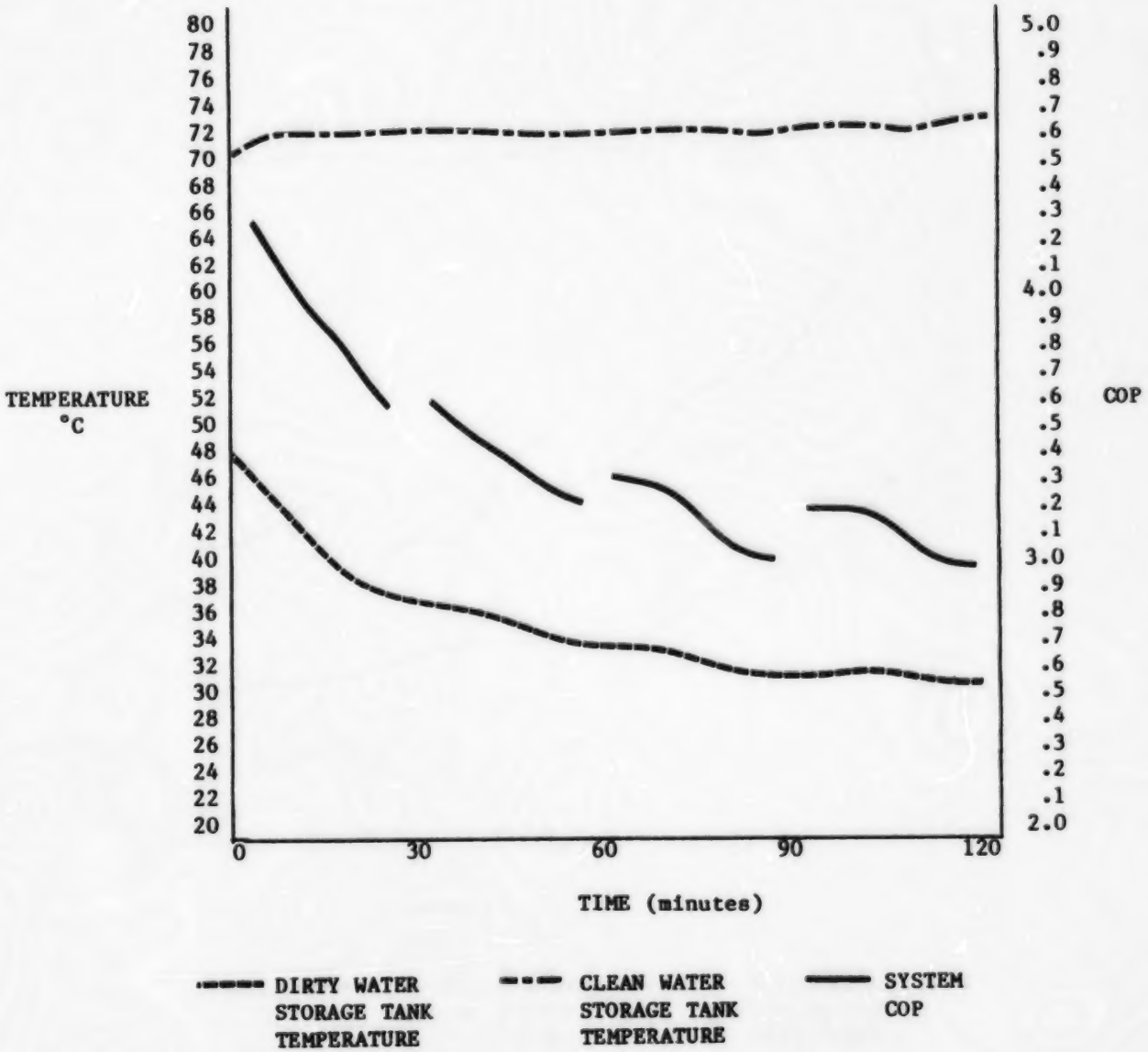
80 series Heat Pump with heat exchanger efficiency of 50%
100% utilization
TINT 22°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



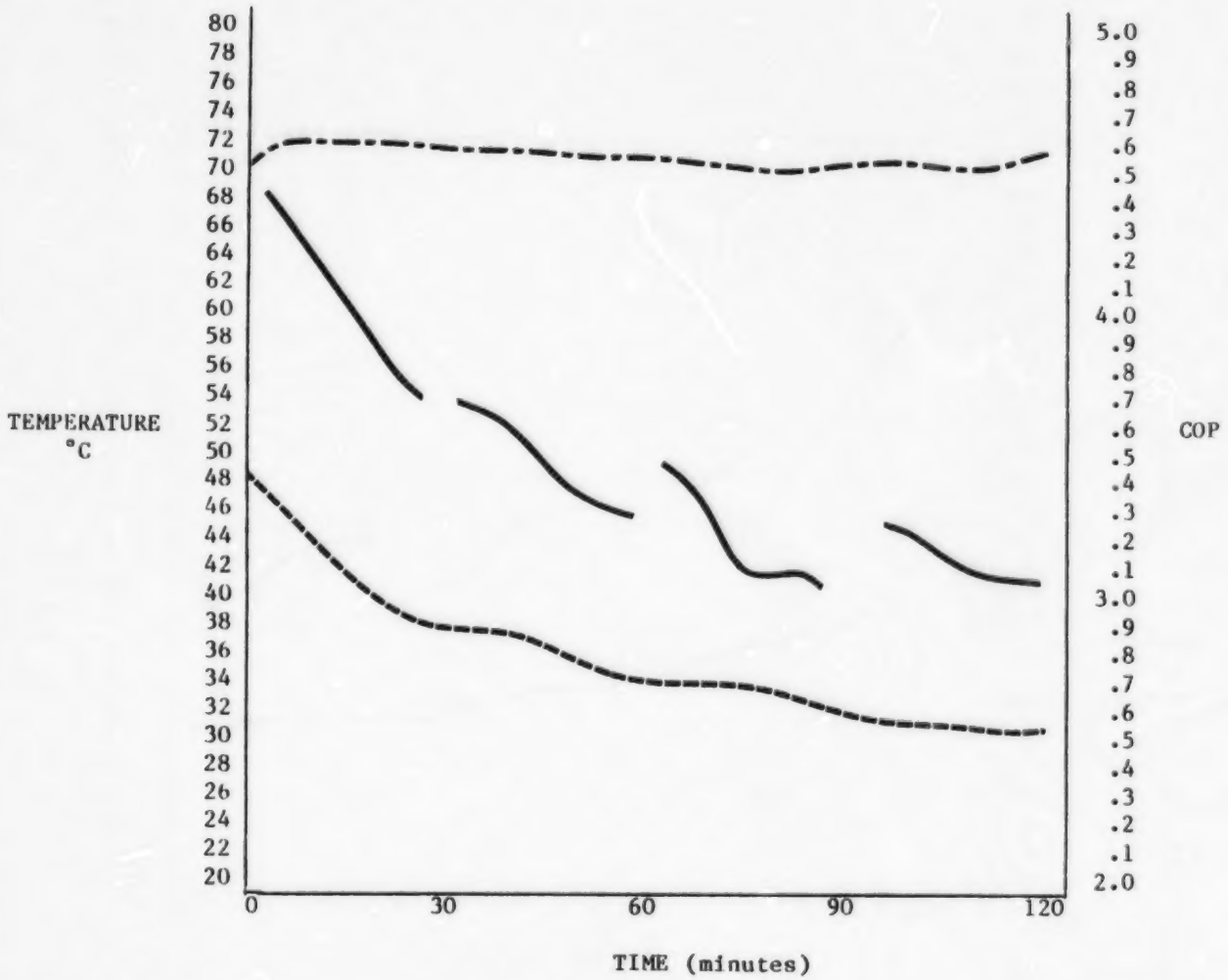
80 series Heat Pump with heat exchanger efficiency of 40%
100% utilization
TINT 22°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



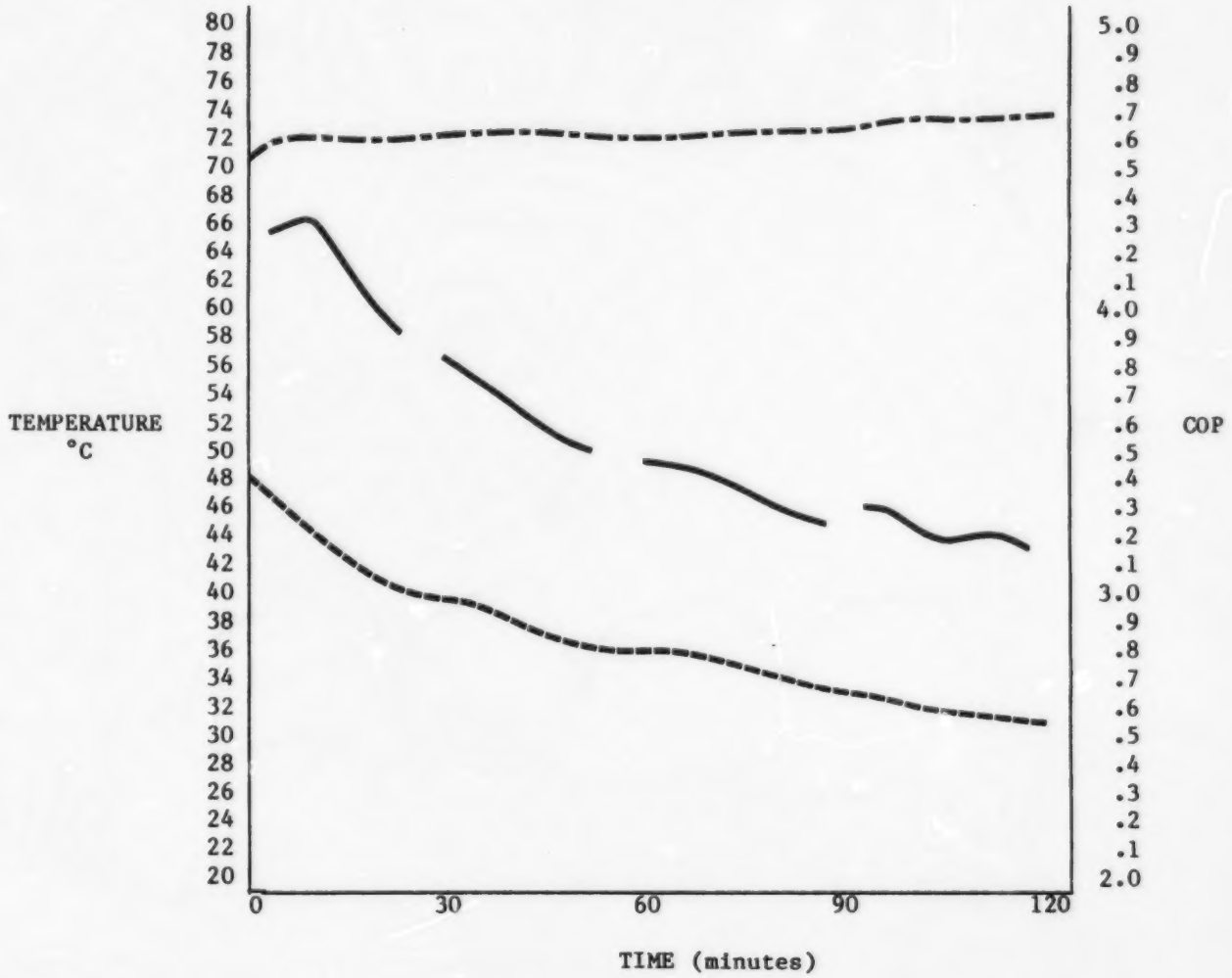
80 series Heat Pump with heat exchanger efficiency of 40%
75% utilization
TINT 48°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



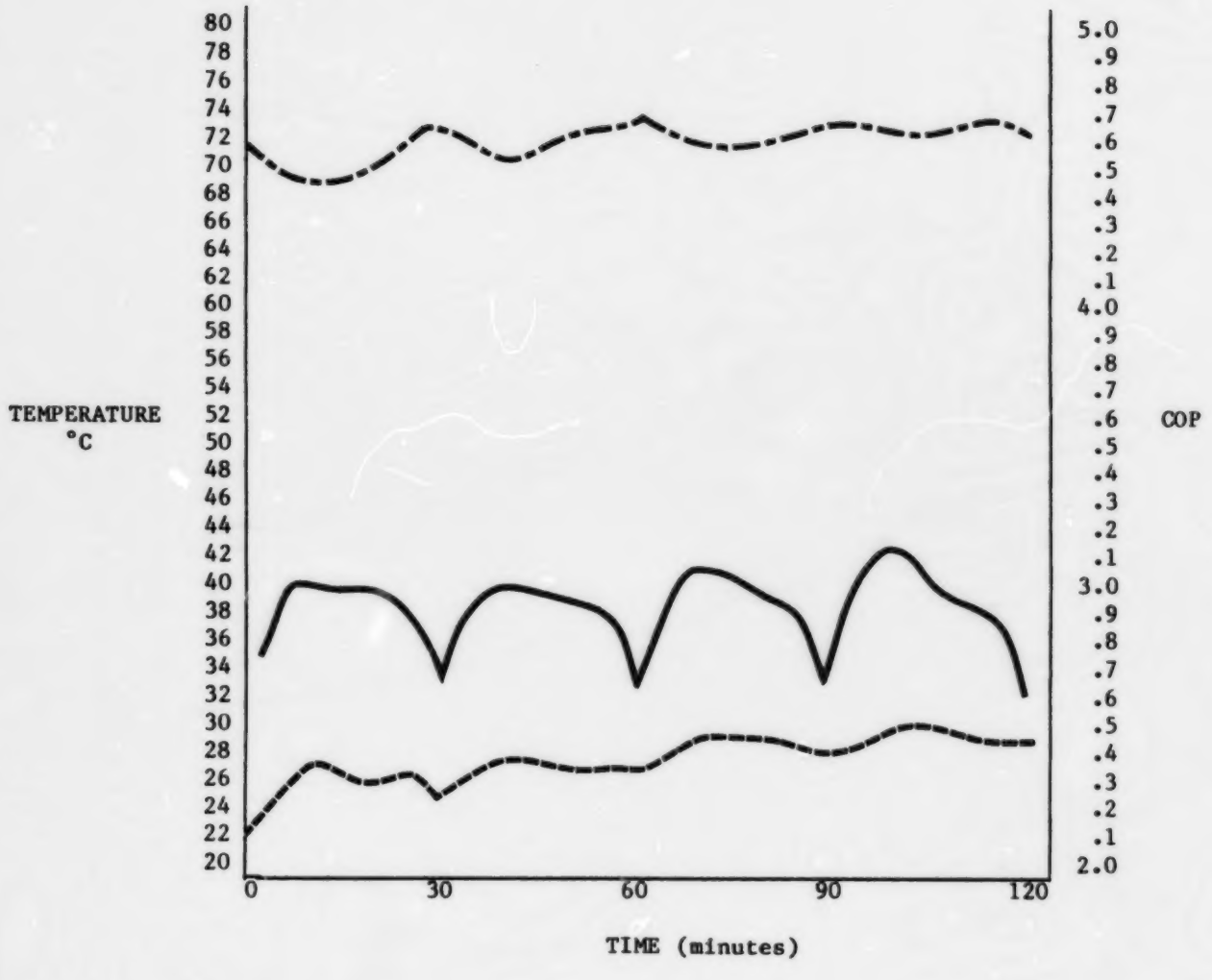
80 series Heat Pump with heat exchanger efficiency of 50%
75% utilization
TINT 48°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



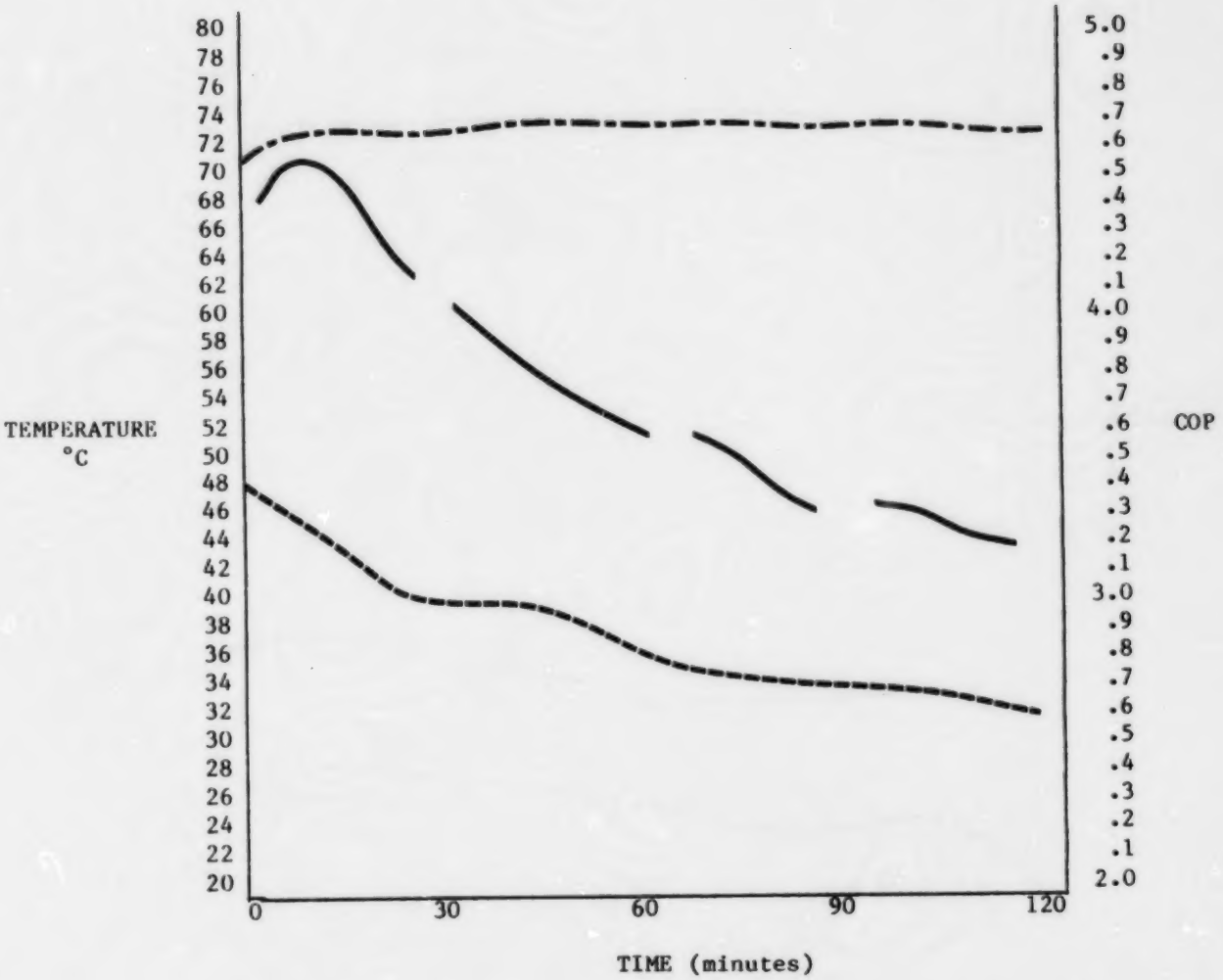
80 series Heat Pump with heat exchanger efficiency of 50%
50% utilization
TINT 48°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



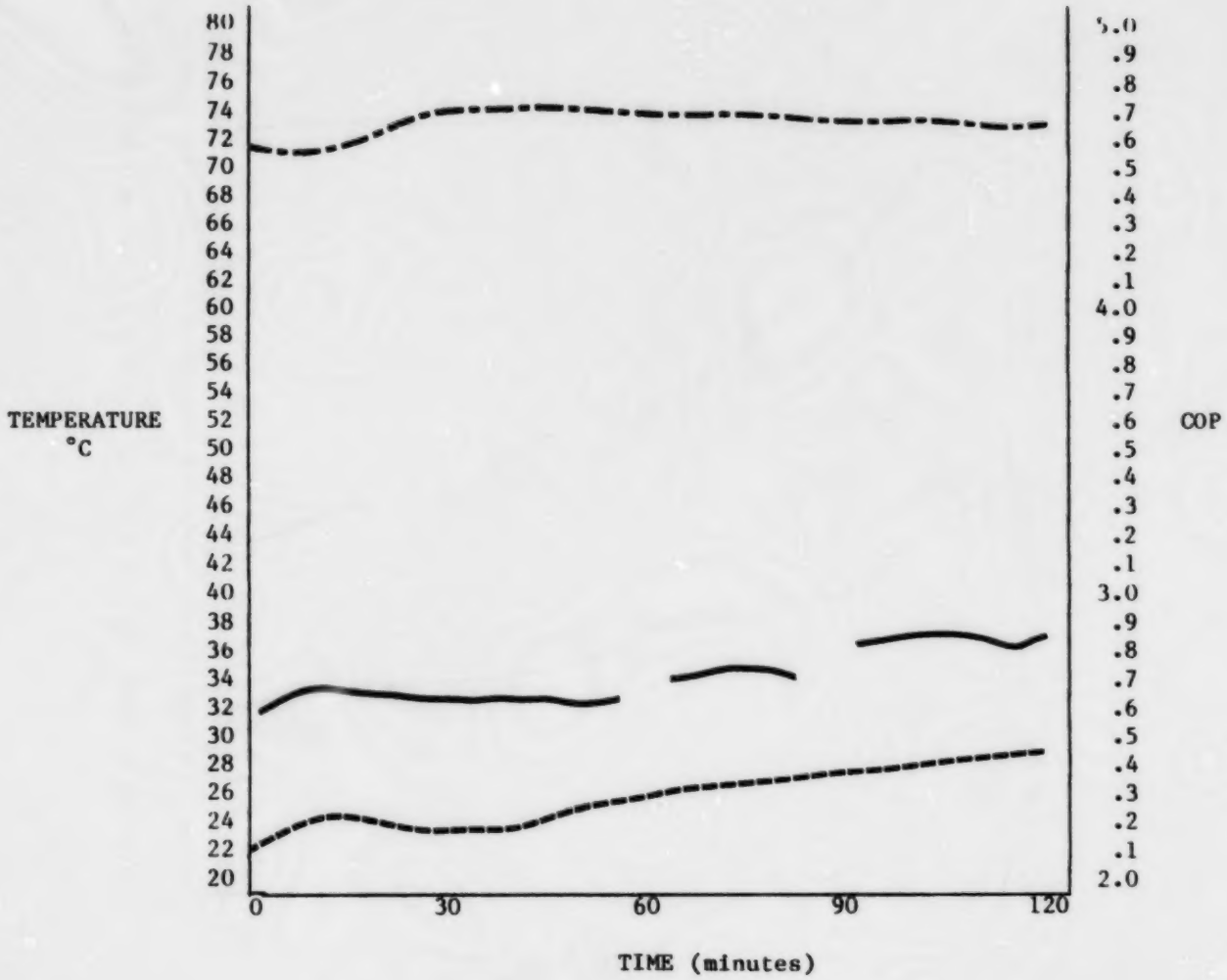
80 series Heat Pump with heat exchanger efficiency of 60%
75% utilization
TINT 22°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



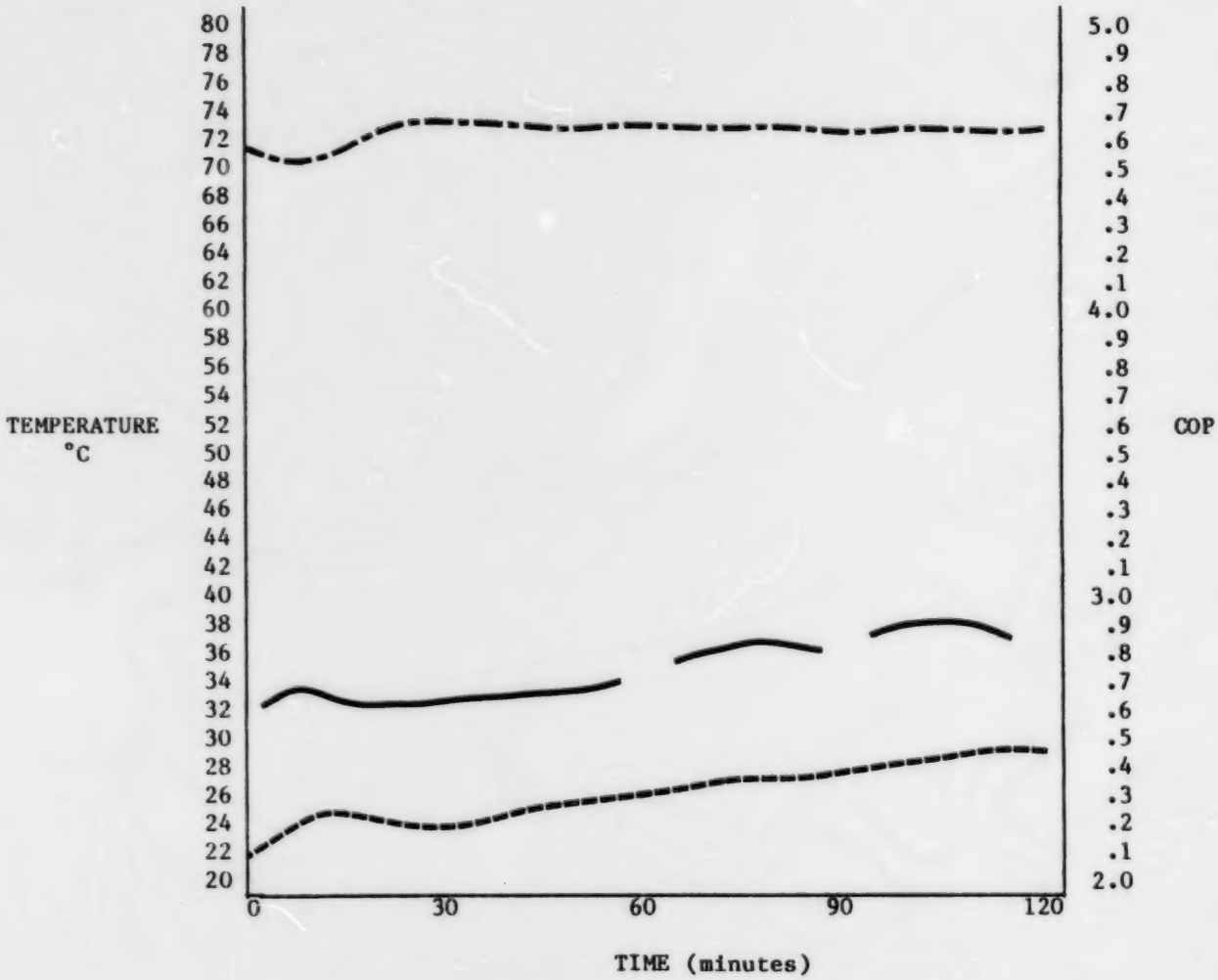
80 series Heat Pump with heat exchanger efficiency of 60%
50% utilization
TINT 48°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



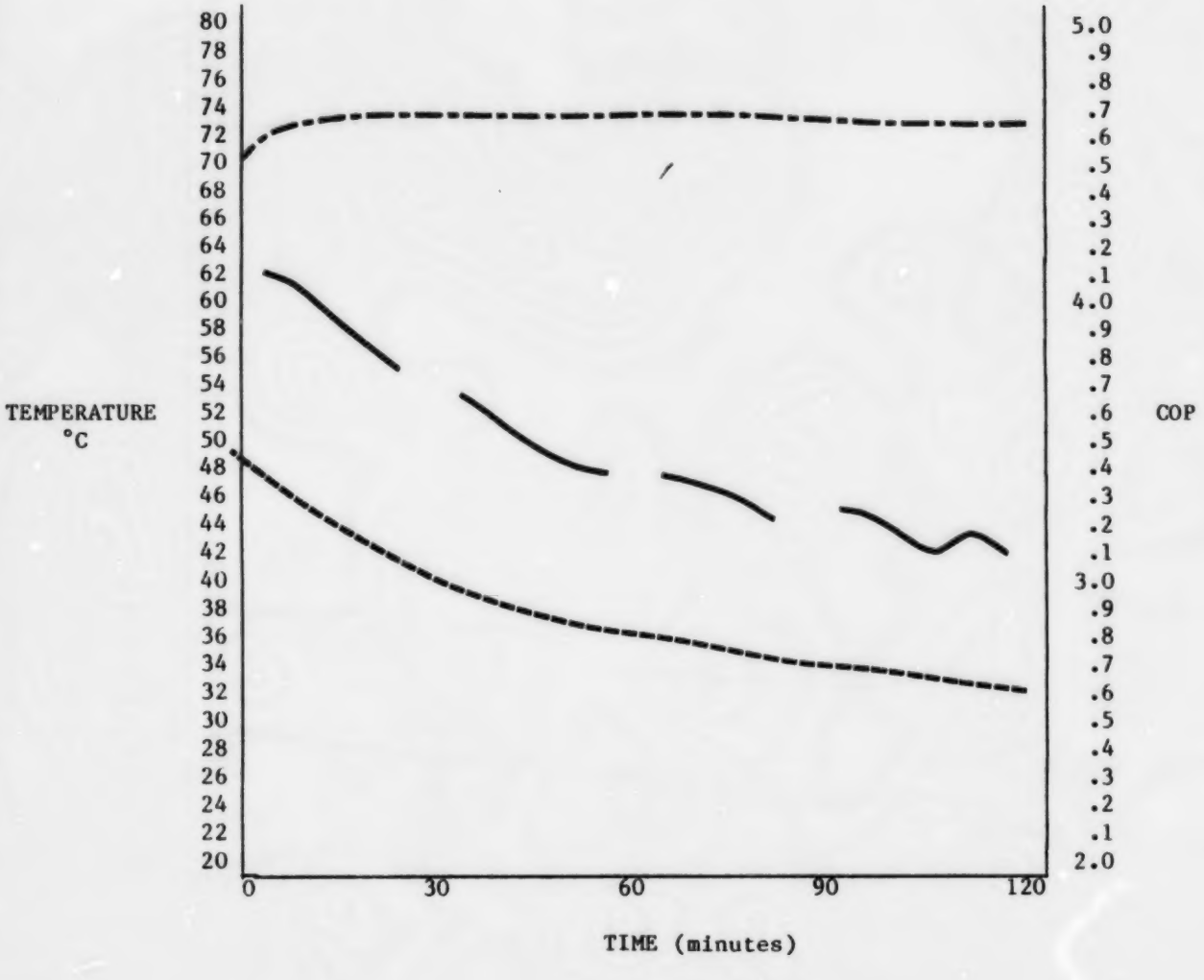
80 series Heat Pump with heat exchanger efficiency of 40%
50% utilization
TINT 22°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



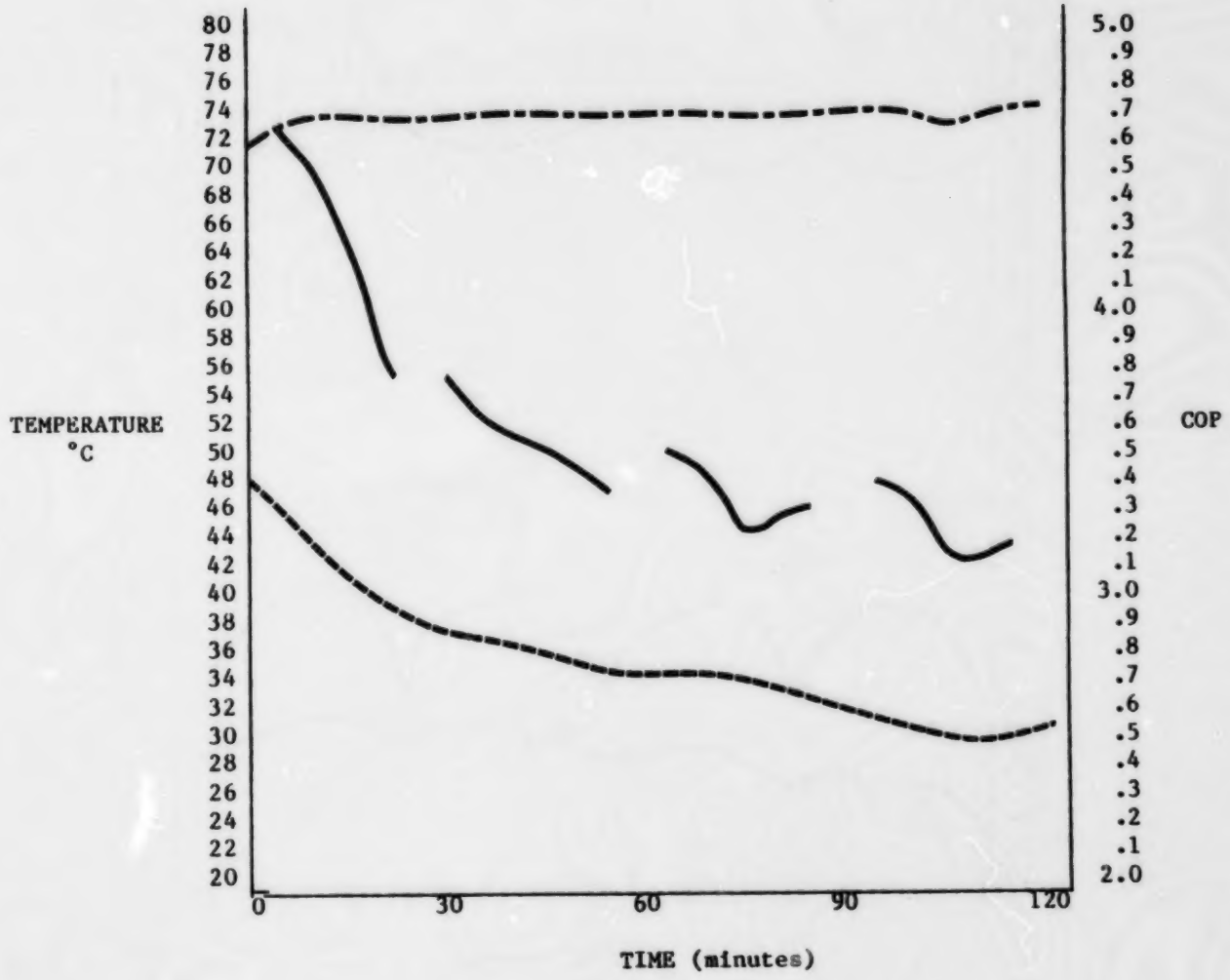
80 series Heat Pump with heat exchanger efficiency of 50%
50% utilization
TINT 22°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



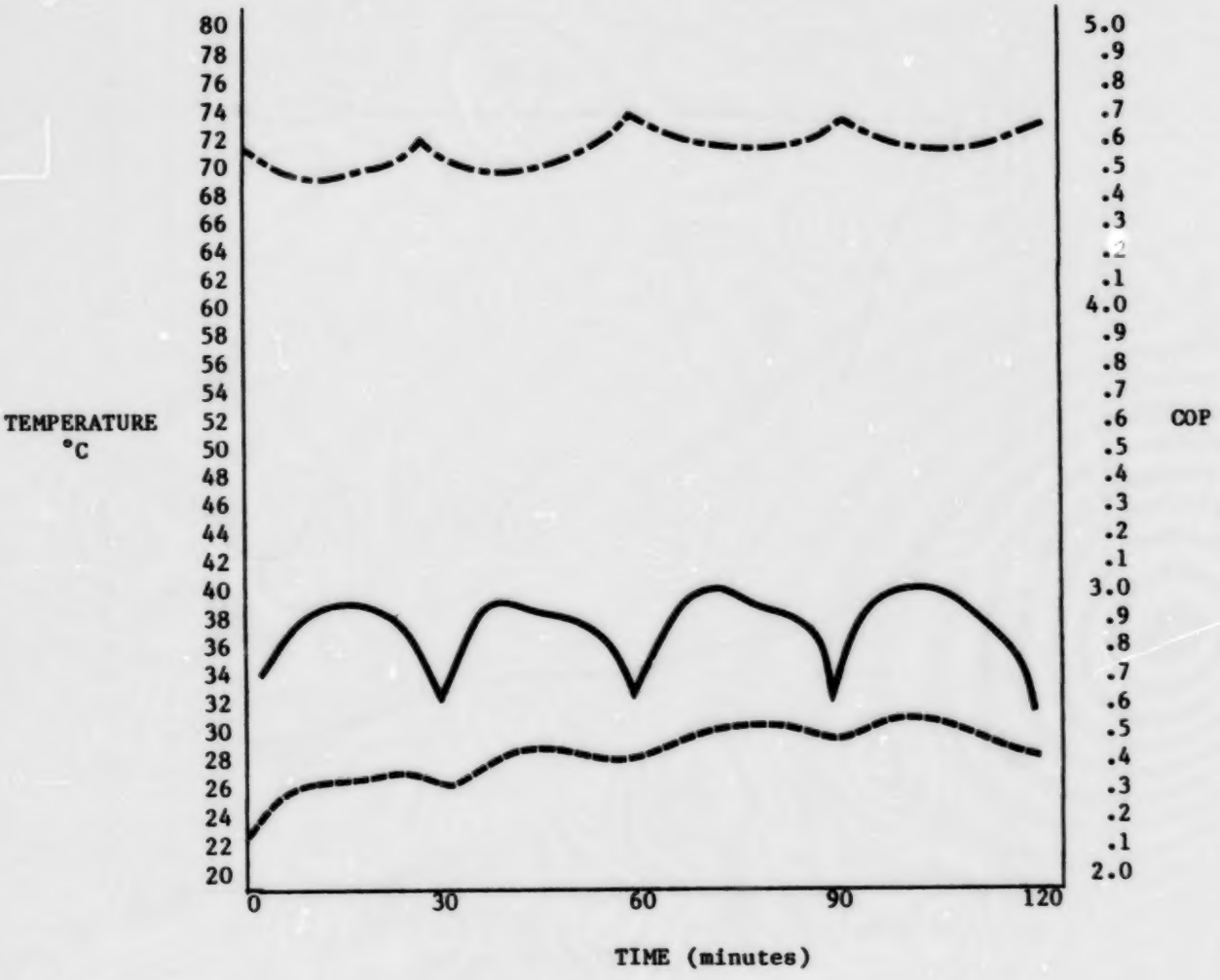
80 series Heat Pump with heat exchanger efficiency of 40%
50% utilization
TINT 48°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



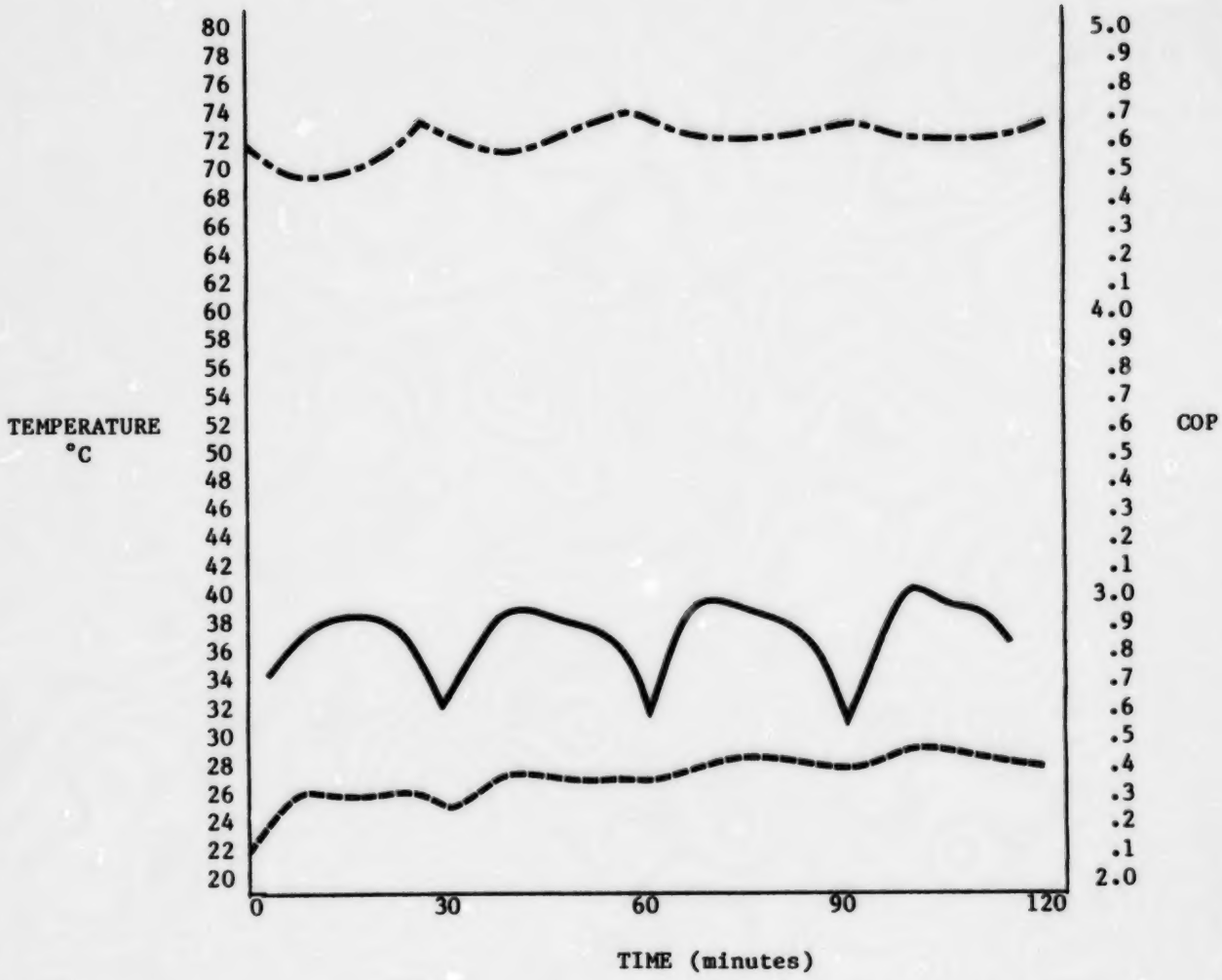
80 series Heat Pump with heat exchanger efficiency of 60%
75% utilization
TINT 48°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



80 series Heat Pump with heat exchanger efficiency of 40%
75% utilization
TINT 22°

GRAPH OF TEMPERATURES AND COP WITH RESPECT TO TIME



80 series Heat Pump with heat exchanger efficiency of 50%
75% utilization
TINT 22°

APPENDIX D

DERIVATION OF THE GENERALIZED BECC ECONOMIC MODEL

DERIVATION OF THE GENERALIZED BECC ECONOMIC MODEL

NOMENCLATURE

C_p = specific heat of water

C^* = escalated energy cost

C_f^* = escalated fuel oil cost, $/litre

C_e^* = escalated electricity cost, $/kwhr.

C_{fo} = fuel oil cost in base year, $/litre

C_{eo} = electricity cost in base year, $/kwhr.

C_o = energy cost in base year

COP = total system coefficient of performance

D = number of days of operation per year; 300 days

d_r = depreciation rate for tax purposes

e = escalation rate (in terms of growth relative to constant $)

e_e = escalation rate, electricity cost

e_f = escalation rate, fuel oil cost

E_{HPK} = annual heat pump system electricity cost, $

F_{BK} = boiler fuel oil cost

H_V = heating value, fuel oil

i = company's projected average rate of return on invested capital (or discount rate)

I = initial investment

i_m = minimum required return on investment for high-risk venture

Iw = working capital

L = number of full load cycles per day

$Lmax$ = maximum number of full load cycles per day

M_{HPK} = heat pump system annual maintenance cost

N = plant life, years

N_B = boiler thermal efficiency

PF = Power to heat ratio; boiler

Q = hot water demand per load cycle

R_K = gross untaxed income (revenue) $/year

r = years allowed for depreciation for tax purposes

Sa = salvage value at end of the n^{th} year

TCI = investment tax credit

t = income tax rate

U = utilization or load factor

W = venture worth, $

T = temperature rise, °F

subscripts: f = fuel oil
e = electricity
K = years

VENTURE WORTH = (GROSS UNTAXED INCOME) + (DISCOUNTED TAX CREDIT FOR DEPRECIATION) - (PRESENT WORTH OF INCREMENTAL RETURN ON TOTAL INVESTMENT) - (INVESTMENT) + (TAX CREDIT) - (PRESENT WORTH OF COST OF WORKING CAPITAL) + (PRESENT WORTH OF SALVAGE)

Thus,

$$W = \sum_{K=1}^{N} \frac{(1-t)R_K}{(1+i)^K} + \sum_{k=1}^{r} \frac{d_r t I}{(1+i)^K} - \left\{\frac{(1+i)^N - 1}{i(1+i)^N}\right\} (i_m - i)(I + Iw) - I + TCI$$

$$- \frac{(1+i)^N - 1}{(1+i)^N} Iw + \frac{1-t}{(I+i)^N} Sa$$

FOR $W = 0$, $I = I_{BE}$

FURTHERMORE, Sa is small at the end of plant life and no change in Iw is expected for operation; thus set Iw and Sa to zero

$$I_{BE} = \frac{\sum_{K=1}^{N} \frac{(1-t)R_K}{(1+i)^K}}{1 + \left(\left\{\frac{(1+i)^N - 1}{i(1+i)^N}\right\} (i_m - i)\right) - TCI - \sum_{K=1}^{r} \frac{d_r t}{(1+i)^K}}$$

Under these conditions, I_{BE} = BECC. Any off-oil taxable grants which may, from time to time, be applicable are included in the first years taxable income, R_1. All terms exclusive of R_K are economic in function and are present independently of system performance. Revenue, R_K, is a function of system performance and load factor.

We now refine the relationship for R_K in terms applicable to the present hot water heating system in operation at Betty Bute and the proposed heat pump waste heat recovery system.

REVENUE, R_K

(1) ANNUAL FUEL OIL COST; boiler, F_{BK}

$$F_{BK} = \frac{L\ Q\ D\ C_p \Delta T}{N_B\ H_V}\ C_F^*$$

L = number of full load cycles per day
Q = hot water demand per load cycle = 500 lbs.
D = number of days of operation per year = 300
Cp = specific heat, water

T = temperature rise, °F = 100

N_B = boiler efficiency = 65%

Hv = fuel oil heating value, Btu/litre = 41,000

C_f^* = escalated fuel oil cost, $/litre

thus

$$F_{BK} = 563.98\ LC_f^*$$

(2) ANNUAL ELECTRICITY COST; Boiler, E_{BK}

KW = 3412 Btuh (conversion factor)

PF = power to heat ratio; for boiler; PF = 0.07

Ce* = escalated electricity cost, $/kWhr

$$E_{BK} = \frac{L\ Q\ D\ Cp\ \Delta T}{N_B\ (3412)}\ PF.C_e^*$$

Thus

$$E_{BK} = 308.35\ LC_e^*$$

(3) ANNUAL ELECTRICITY COST; heat pump system, E_{HPK}

$$E_{HPK} = \frac{L\ Q\ D\ C_p\ \Delta T}{COP\ \ (3412)}\ C_e^*$$

COP = Coefficient of Performance, heat pump system

thus

$$E_{HPK} = \frac{4405.04}{COP}\ LC_e^*$$

(4) Annual Maintenance Cost

Maintenance functions presently supported by Betty Brite's existing operation are not expected to change with the conversion to the proposed heat pump system. Thus labour costs will remain the same. However, the conversion will result in the removal of the mixing valve which presently involves annual maintenance of $1500. It is furthermore expected that heat pump system reliability is comparable to that of commercial refrigeration systems. However, to be conservative in this matter, HPI proposes that a capital replacement fund of $200/year be generated to replace unexpected equipment failures over and above that currently used for the oil-fired hot water boiler.

Thus, the heat pump system Annual Maintenance Cost, M_{HPK}, is -$1,300/yr (maintenance actually a revenue in this case and independent of L).

In summary,

$$R_K = F_{BK} + E_{BK} - H_{PK} - M_{HPK}$$

$$= 563.98LC_f^* + 308.35\ LC_e^* - \frac{4405.04}{COP}\ LC_e^* + 1300$$

letting U = utilization (load factor)

$$= \frac{L}{Lmax}\ ,$$ where Lmax = 15 (i.e. 2 loads per hour for 7.5 hrs per day)

$$L = 15U$$

and

$$R_K = 8459.7UC_f^* + 4625.3\ UC_e^* - \frac{66075.6}{COP}\ UC_e^* + 1300$$

ENERGY COST ESCALATION FACTORS

$$C^* = Co\ (1 + e)^K$$

where C^* = escalated energy cost

Co = energy cost in base year

e = annual escalation rate

$C_f^* = C_{fo}\ (1 + e_f)^K$ f subscript = fuel oil

$C_e^* = C_{eo}\ (1 + e_e)^K$ e subscript = electricity

GENERALIZED BECC ECONOMIC MODEL

The generalized economic model thus becomes

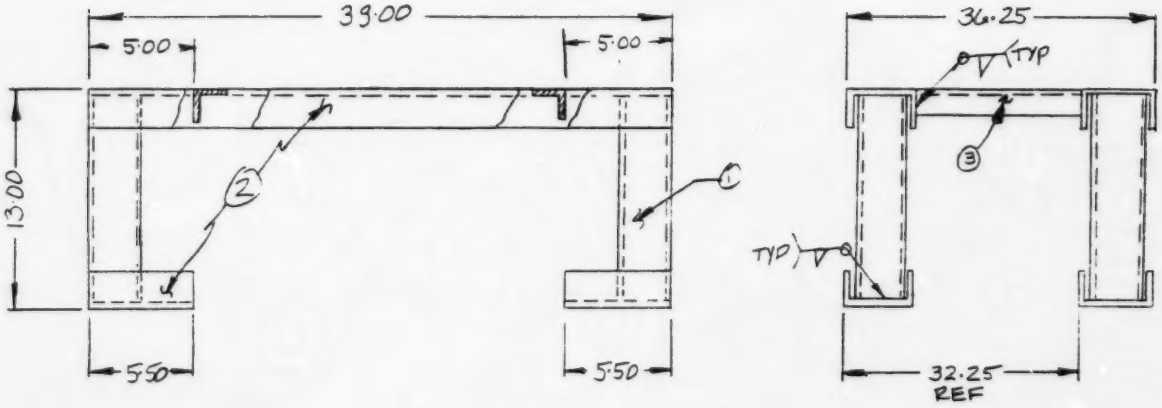
$$BECC = \frac{\sum_{K=1}^{N} \frac{(1-t)^{R_K}}{(1+i)^K}}{1 + \left(\left\{ \frac{(1+i)^N - 1}{i(1+i)^N} \right\} (i_m - i) \right) - TCI - \sum_{K=1}^{r} \frac{d_r t}{(1+i)^K}}$$

where

$$R_K = 8460\ UC_{fo}\ (1 + e_f)^K + UCe_o\ (1 + e_e)^K \left(4625 - \frac{66076}{COP}\right) + 1300$$

APPENDIX E

PROTOTYPE WASTE WATER STORAGE TANK



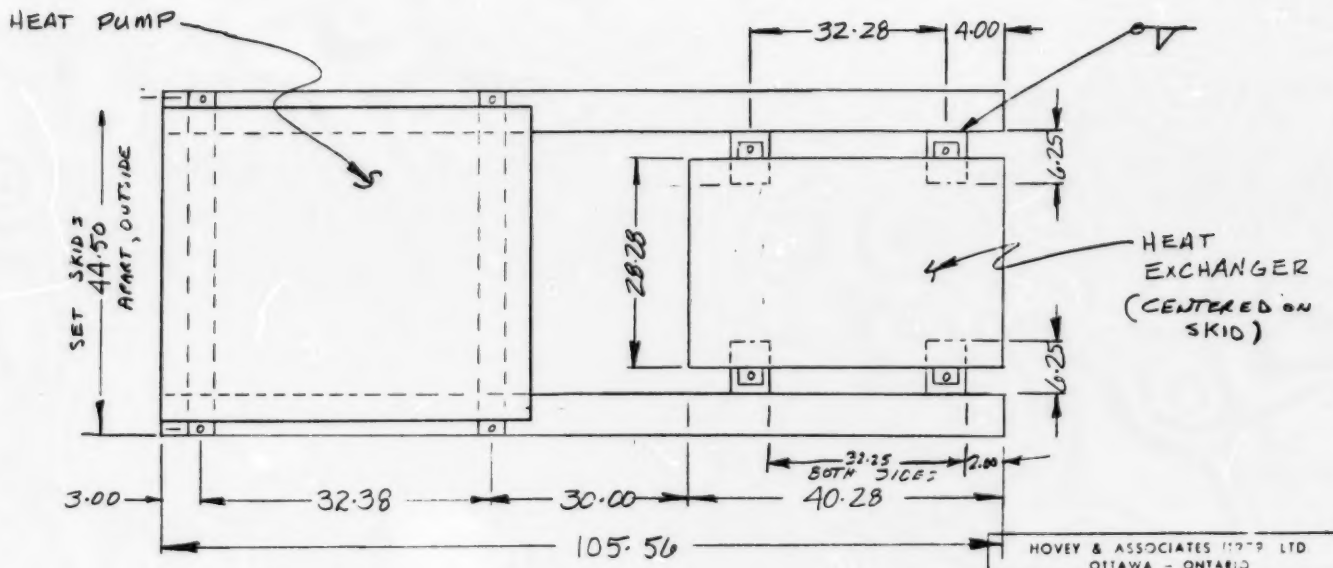
MAT'L ① $2\frac{1}{2} \times 2\frac{1}{2} \times \frac{1}{8}$ ($\frac{3}{16}$) ST TUBING

② 4 × 1.65 × .25 THK CHANNEL

③ $1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{8}$ ST ANGLE

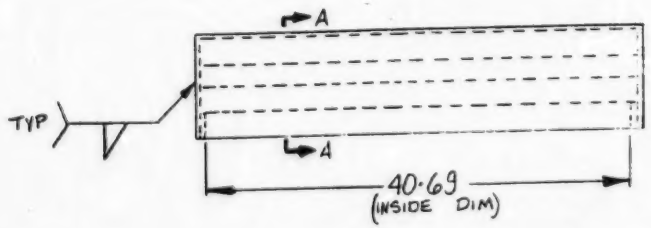
NOTE: PAINT HAC GREY

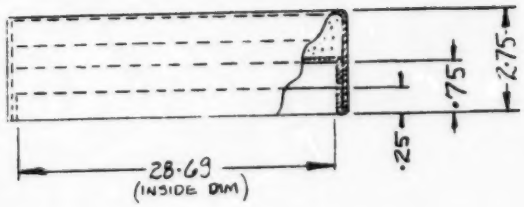
HOVEY & ASSOCIATES (1979) LTD. OTTAWA - ONTARIO		
TITLE TANK STAND		
FOR HEAT EXCHANGER		
SCALE .	DWN RB	DWG. NO 2012-01
DATE FEB 04	CHKD. —	
QTY. 1	APP'D.	SHT 1 OF 1

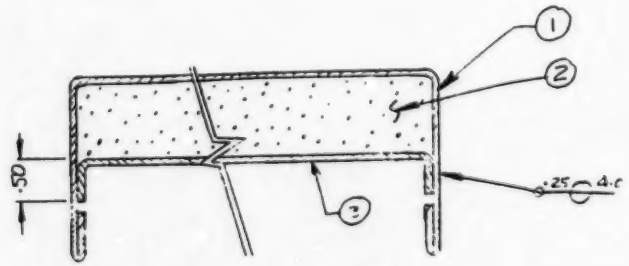


MAT'L: 4 × 1·65 × ·25 THK CHANNEL 2 pcs 105·56 lg
4 pcs 6·25 lg
PAINT GREY (HAC)

HOVEY & ASSOCIATES (1979) LTD OTTAWA – ONTARIO		
TITLE SKID		
FOR HEAT EXCHANGER		
SCALE —	DWN RB	DWG NO 2012-010
DATE Feb 1	CHKD. KH	
QTY 1	APP'D	SHT OF





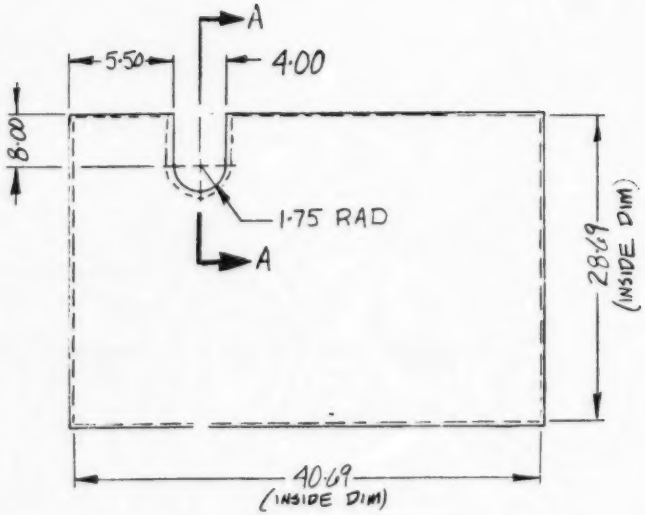


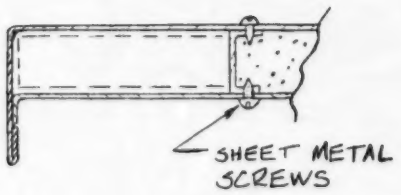
SECTION A-A

MAT'L: ① 14 GA SST
② INSULATION
③ 20 GA SST

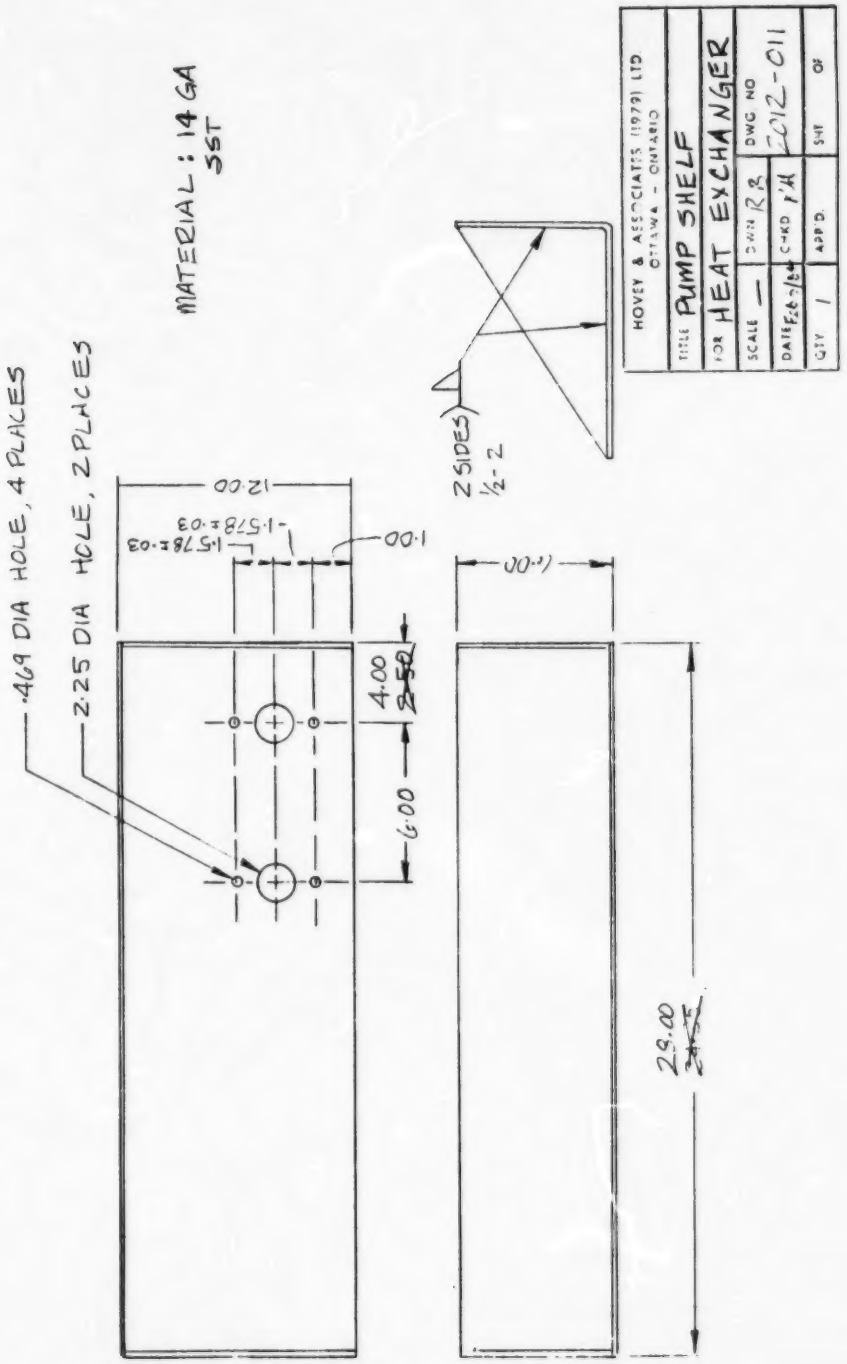
NOTE:- REMOVE ALL SHARP EDGES
-MIN RAD ON ALL BENDS

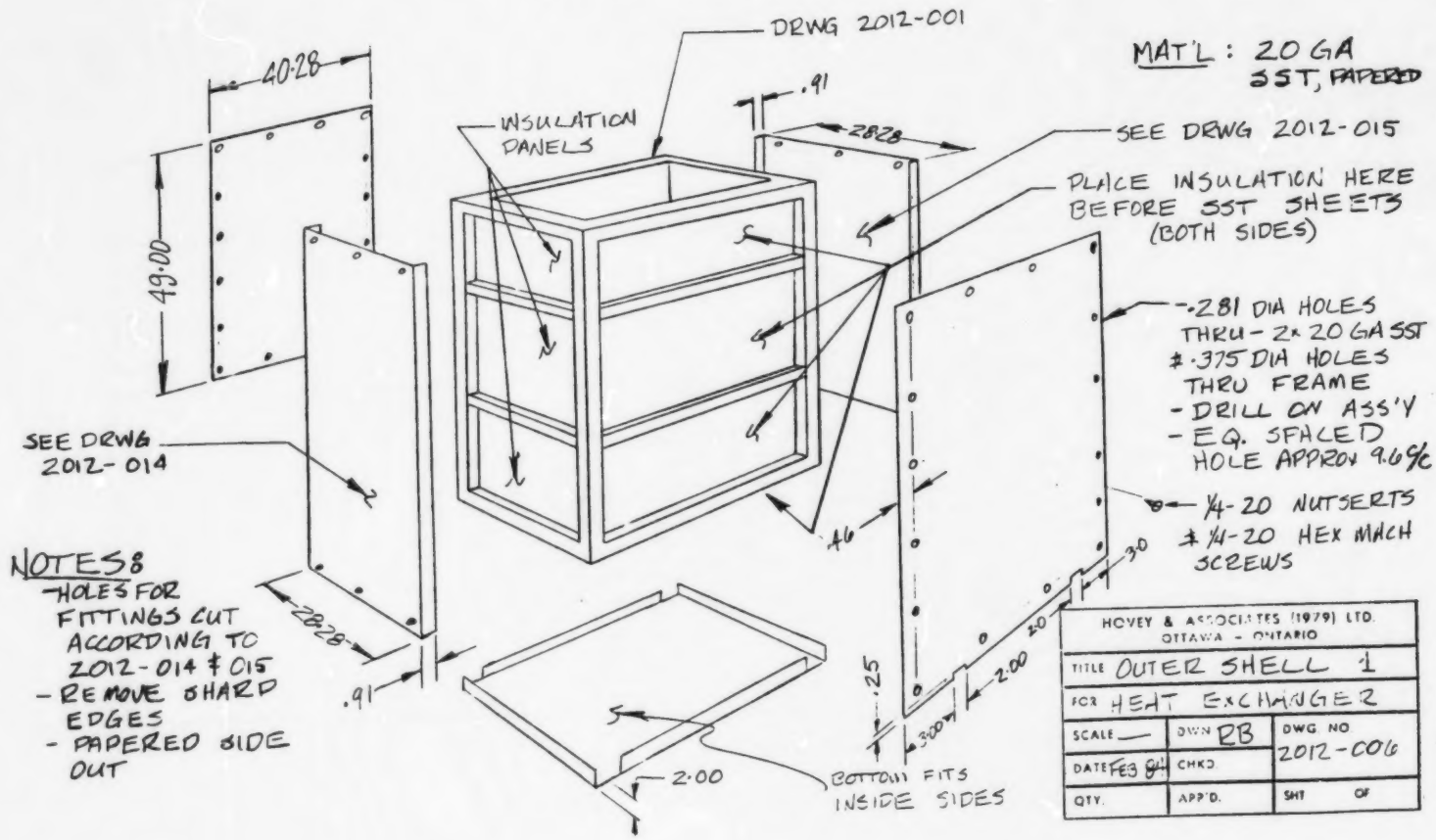
HOVEY & ASSOCIATES (1979) LTD. OTTAWA - ONTARIO		
TITLE REMOVABLE TOP		
FOR HEAT EXCHANGER		
SCALE —	D.WN RB	DWG. NO 202-005
DATE FEB 84	C.HKD.	
QTY.	APP'D.	SHT 1 OF 2





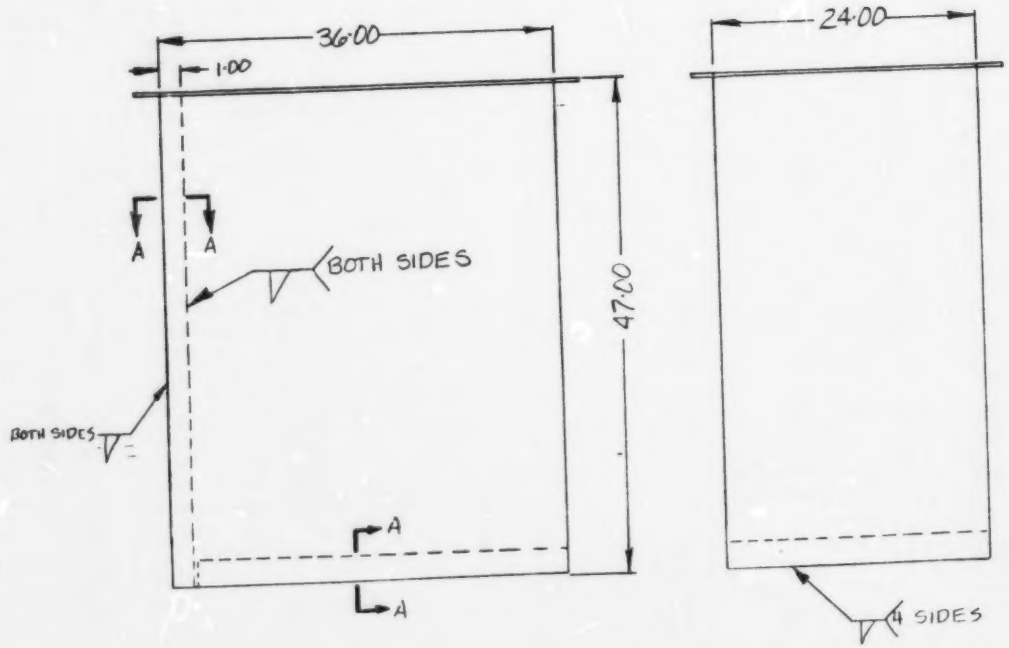
HOVEY & ASSOCIATES (1979) LTD. OTTAWA - ONTARIO		
TITLE TOP VIEW OF LID		
FOR HEAT EXCHANGER		
SCALE —	DWN RB	DWG NO 2012-005
DATE	CHKD	
QTY 1	APP'D	SHT 2 OF 2


.469 DIA HOLE, 4 PLACES
2.25 DIA HOLE, 2 PLACES
MATERIAL: 14 GA SST
12.00
1.578±.03
1.578±.03
1.00
4.00
2.50
6.00
1.00
29.00
2 SIDES ½-2
HOVEY & ASSOCIATES (1979) LTD
OTTAWA - ONTARIO
TITLE PUMP SHELF FOR HEAT EXCHANGER
SCALE —
DWN RR
DWG NO 2012-011
DATE Feb/84
CHKD PH
QTY 1
APP'D.
SHT OF



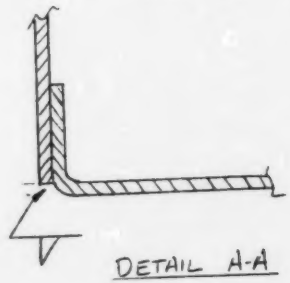
NOTES:
- HOLES FOR FITTINGS CUT ACCORDING TO 2012-014 & 015
- REMOVE SHARP EDGES
- PAPERED SIDE OUT

HOVEY & ASSOCIATES (1979) LTD. OTTAWA - ONTARIO		
TITLE OUTER SHELL 1		
FOR HEAT EXCHANGER		
SCALE —	DWN RB	DWG NO 2012-C06
DATE FEB 84	CHK'D	
QTY.	APP'D.	SHT OF



CAPACITY = 594651 cm³
= 130 imp gals

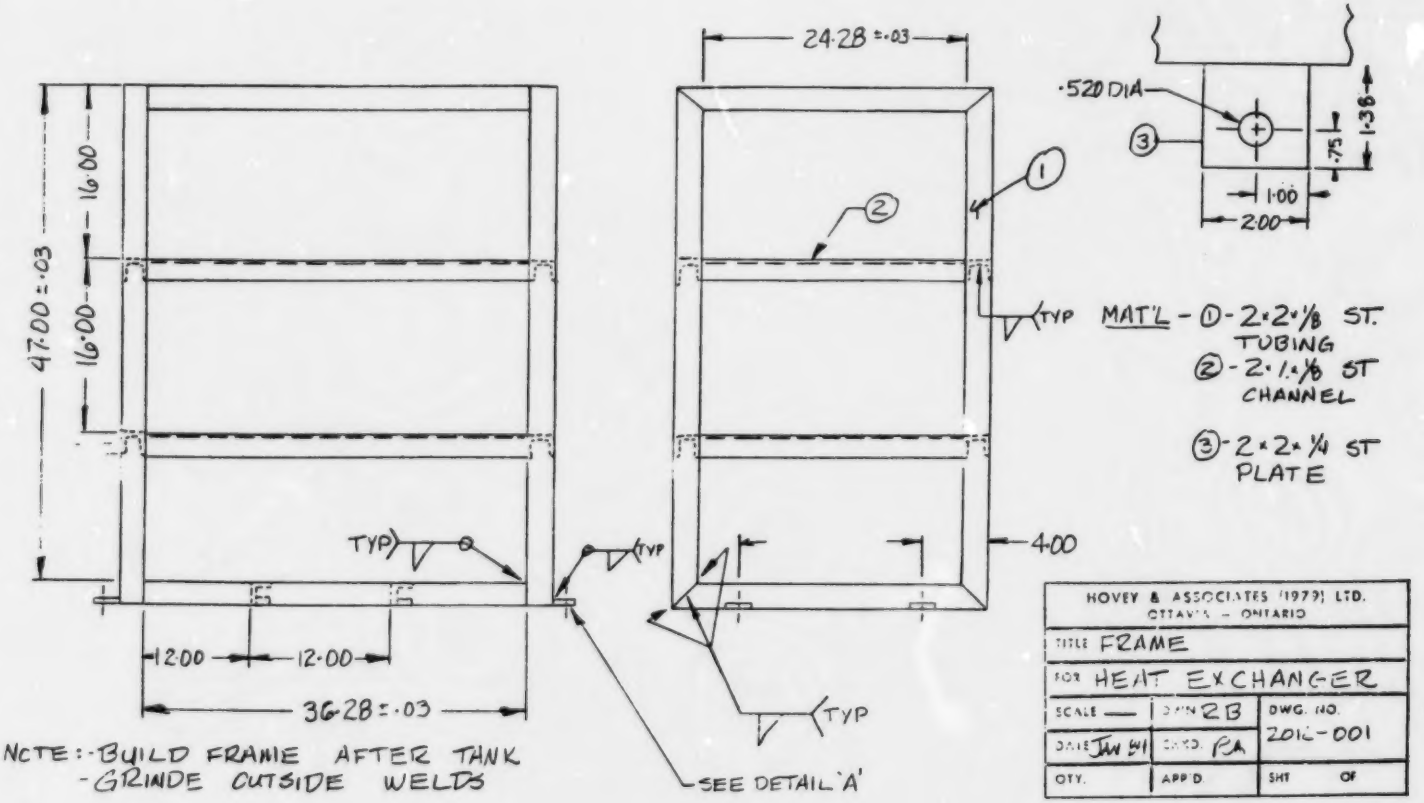
MATL: 14 GA S.ST



DETAIL A-A

NOTE: - REMOVE SHARP EDGES & CORNERS
- INSIDE DIMENSIONS GIVEN
- ±·03 ON ALL DIMENSIONS
- BUILD TANK BEFORE FRAME

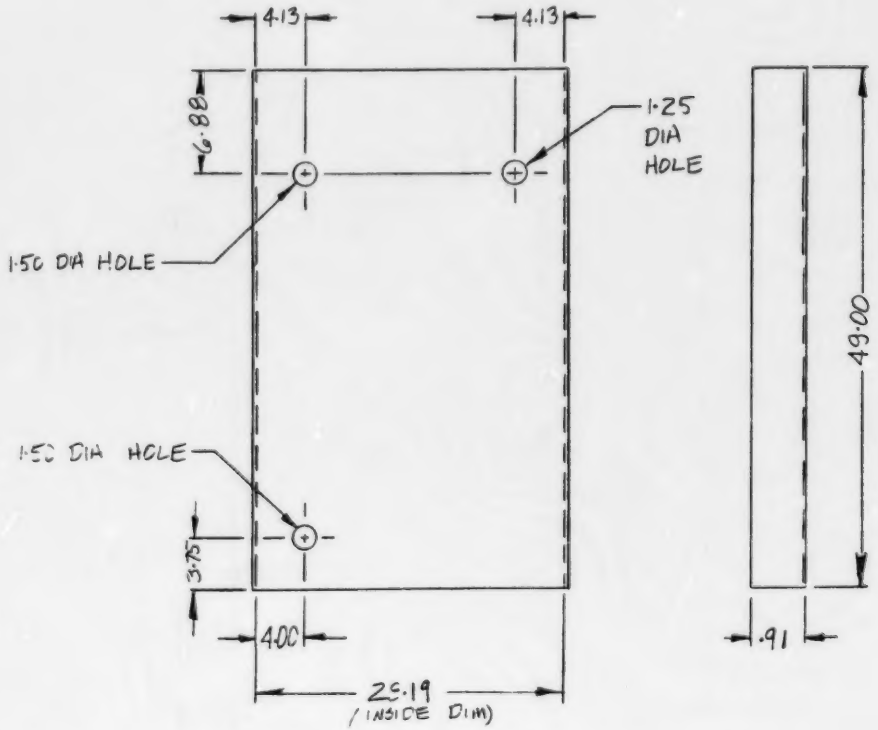
HOVEY & ASSOCIATES (1979) LTD. OTTAWA - ONTARIO		
TITLE INNER SHELL		
FOR HEAT EXCHANGER		
SCALE —	DRN RB	DWG NO 202-002
DATE JAN 84	CHKD RA	
QTY	APP'D	SHT OF



MAT'L - ① - 2 × 2 × 1/8 ST. TUBING
② - 2 × 1 × 1/8 ST CHANNEL
③ - 2 × 2 × 1/4 ST PLATE

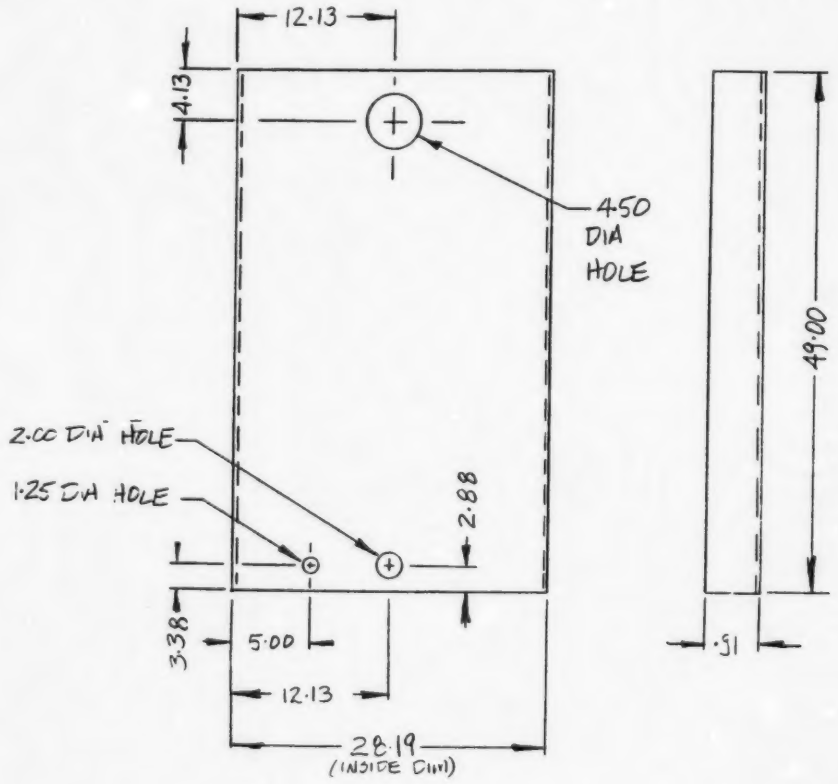
NOTE: - BUILD FRAME AFTER TANK
- GRIND OUTSIDE WELDS

HOVEY & ASSOCIATES (1979) LTD. OTTAWA - ONTARIO		
TITLE FRAME		
FOR HEAT EXCHANGER		
SCALE —	DWN ZB	DWG. NO. 201C-001
DATE JAN 81	CHKD. RA	
QTY.	APP'D	SHT OF



MAT'L 20 GA SST

HOVEY & ASSOCIATES (1979) LTD. OTTAWA – ONTARIO		
TITLE OUTER SHELL, RT SIDE		
FOR HEAT EXCHANGER		
SCALE —	DWN R.K.	DWG NO 2012-014
DATE FEB 84	CHKD.	
QTY.	APP'D.	SHT OF

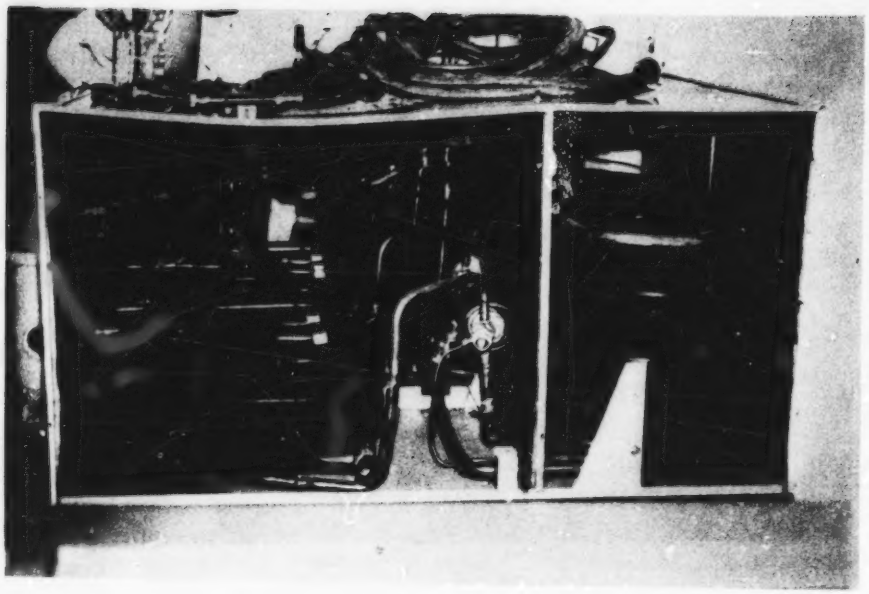


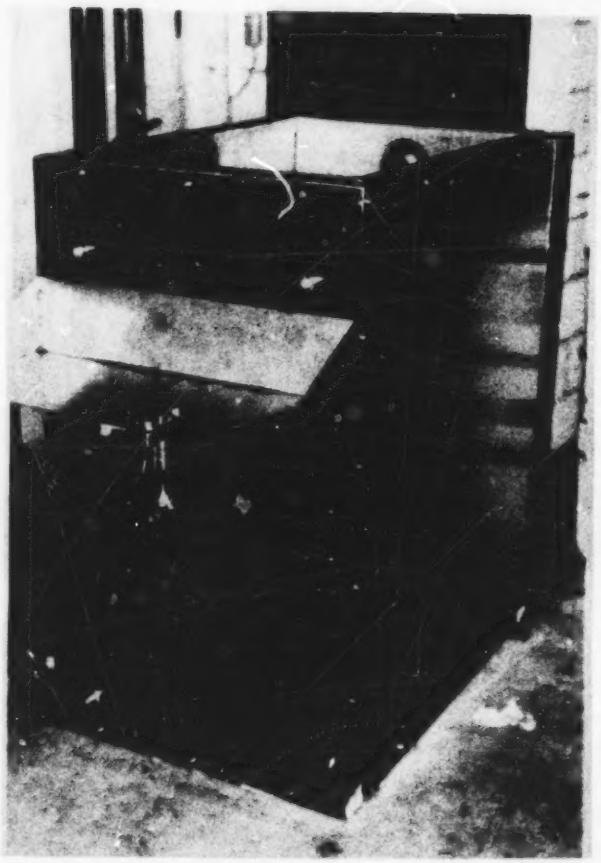
MATL 20 GA SST

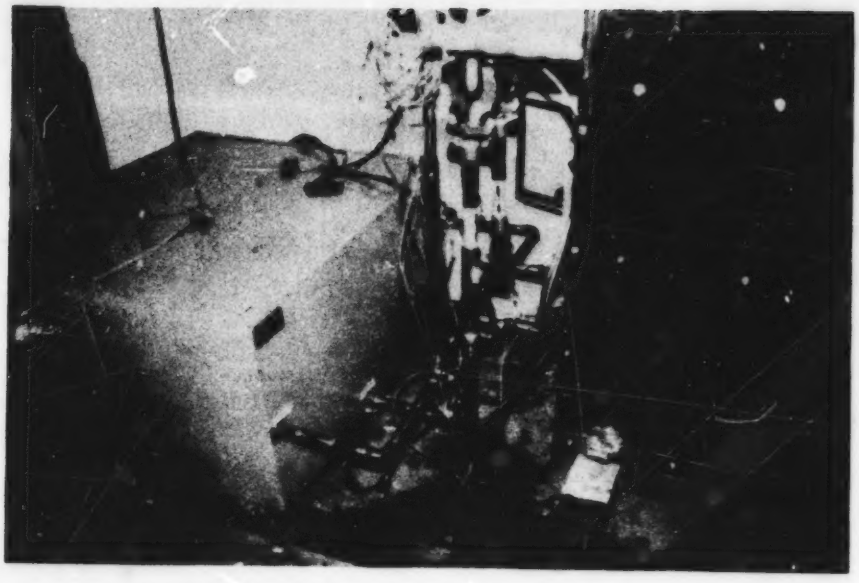
HOVEY & ASSOCIATES (1979) LTD. OTTAWA - ONTARIO		
TITLE OUTER SHELL LT. SIDE		
FOR HEAT EXCHANGER		
SCALE —	DWN RB	DWG. NO. 2012-015
DATE FEB 84	CHKD.	
QTY.	APP'D.	SHT OF

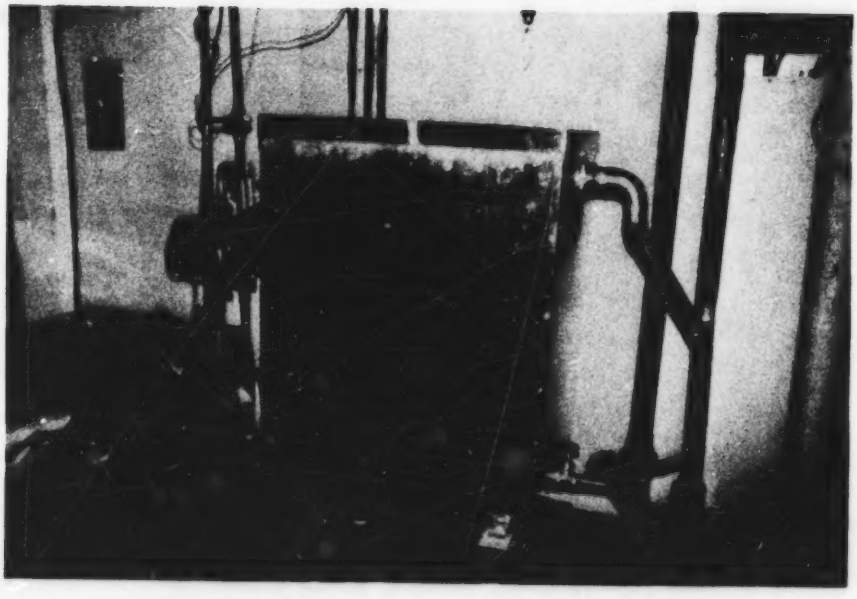
APPENDIX F

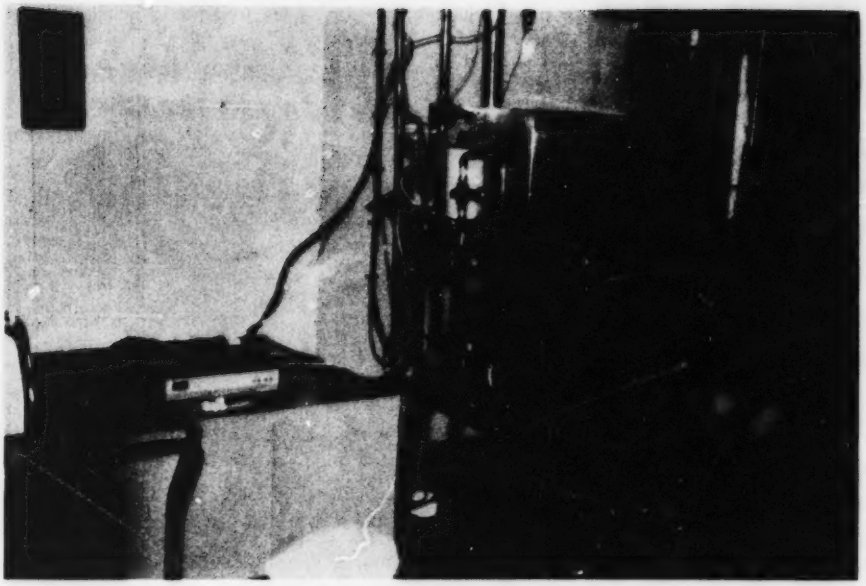
AS-INSTALLED HEAT PUMPS WASTE HEAT RECOVERY SYSTEM



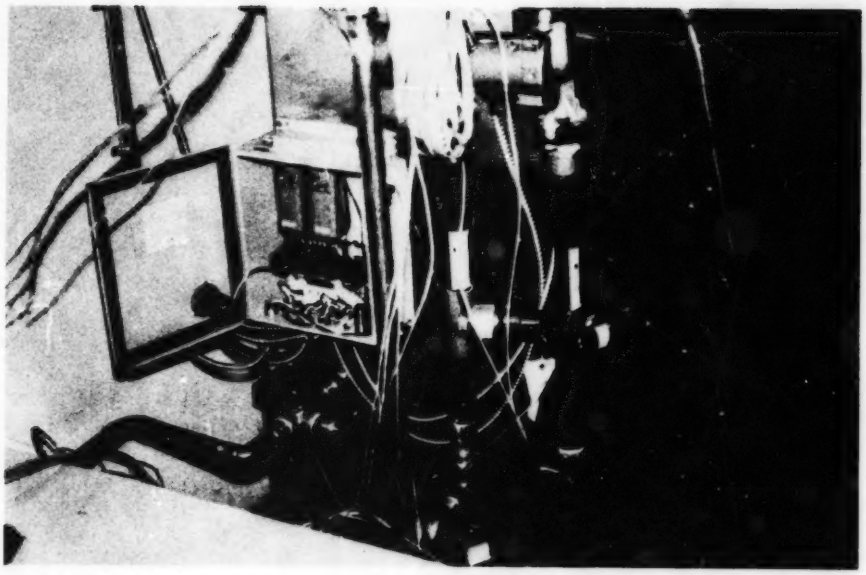








Mirror



APPENDIX G

PROTOTYPE HEAT PUMP WASTE HEAT RECOVERY SYSTEM

SAMPLE DATA

DATA FILES

5.25 inch IBM Compatible floppy diskettes

FILE NAME	SIZE (BYTES)
BB1.DiF	73974
BB2.DiF	5872
BB3.DiF	79428
BB4.DiF	71546
BB5.DiF	74054
BB6.DiF	36324
BB7.DiF	27607
BB8.DiF	76766
BB9.DiF	18517
BB10.DiF	43513

CHANNEL	DESCRIPTION
C1	TIME OF MEASUREMENT .HOUR
C2	WASHING MACHINE - SYSTEM ON IFF READING 1
C3	HOT WATER TANK - SYSTEM ON IFF READING 1
C4	HEAT PUMP SYSTEM- SYSTEM ON IFF READING 1
C5	FRESHWATERFLOW RATE THROUGH CONDENSER .L/MIN
C6	FRESH WATER TEMPERATURE AT CONDENSER INLET .oC
C7	FRESHWATER TEMPERATUREAT CONDENSEROUTLET.oC
C8	WASTE WATER FLOW RATE THROUGH EVAPORATOR .L/MIN
C9	WASTE WATER TEMPERATURE AT EVAPORATOR INLET.oC
C10	WASTE WATER TEMPERATURE AT EVAPORATOR OUTLET.oC
C11	FRESH HOT WATER STORAGE TEMPARATURE .oC

C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11
7.99	1	0	1	36	59.7	65.5	9	25.1	11.7	61.3
8.04	1	0	1	37.8	60.3	66.1	7.2	25.1	12.3	61.8
8.09	0	0	1	37.8	60.7	66.7	7.2	25.1	13.5	62.4
8.14	0	0	1	36	61.0	67.2	9	24.3	12.6	62.6
8.19	1	0	1	37.8	61.2	67.2	5.4	23.9	11.4	62.7
8.24	1	0	1	37.8	61.7	67.6	3.6	23.2	11.0	63.0
8.29	0	0	1	36	62.1	68.2	7.2	22.8	10.6	63.4
8.40	1	0	1	36	57.5	56.0	7.2	22.0	5.66	52.8
8.45	1	0	1	36	59.5	64.5	5.4	21.7	9.81	60.0
8.55	1	0	1	37.8	58.9	63.4	5.4	21.7	9.07	59.0
8.65	1	0	1	36	57.9	62.5	0	22.5	7.83	58.3
8.70	1	0	1	37.8	58.9	64.2	3.6	22.3	8.47	59.9
8.75	1	0	1	37.8	58.5	62.8	0	22.8	9.59	58.7
8.80	1	0	1	36	57.9	59.5	0	24.3	9.99	56.0
8.85	1	0	1	37.8	59.0	64.9	0	26.9	12.9	60.6
8.90	1	0	1	37.8	59.7	65.8	7.2	25.9	12.8	61.4
8.95	1	0	1	37.8	60.6	66.2	14.4	25.9	12.1	61.8
9.00	1	0	1	37.8	60.7	64.7	7.2	25.3	11.8	62.3
9.05	1	0	1	36	60.2	66.4	0	24.3	11.0	62.1
9.10	0	0	1	36	60.6	66.8	5.4	24.2	11.6	62.4
9.20	1	0	1	36	57.8	62.9	7.2	23.1	7.60	58.5
9.25	1	0	1	36	58.9	64.8	7.2	24.2	11.5	60.4
9.30	1	0	1	36	60.1	66.5	7.2	24.2	12.4	61.9
9.45	1	0	1	37.8	58.6	63.9	3.6	25.6	12.2	59.7
9.50	1	0	1	37.8	59.4	65.0	5.4	25.2	12.3	60.8
9.55	1	0	1	37.8	60.0	66.1	0	25.0	12.9	61.8
9.60	0	0	1	37.8	60.4	66.5	0	25.1	12.8	62.3
9.70	1	0	1	37.8	58.0	62.0	3.6	24.5	11.0	57.8
9.75	1	0	1	37.8	58.8	63.7	5.4	24.3	11.2	59.6
9.85	1	0	1	37.8	56.7	61.4	0	25.2	9.98	57.3
9.91	1	0	1	36	58.5	63.6	7.2	24.9	11.3	59.5
9.95	1	0	1	37.8	57.8	61.5	3.6	25.0	9.81	57.5
10.0	1	0	1	36	58.6	63.8	1.8	24.4	10.4	59.5
10.1	1	0	1	36	58.7	56.5	7.2	25.1	10.1	54.0
10.2	1	0	1	37.8	58.8	64.5	0	28.2	13.8	60.2
10.2	1	0	1	36	59.3	65.3	1.8	27.3	13.4	60.9
10.3	1	0	1	37.8	60.0	66.2	0	26.7	12.7	61.7
10.3	1	0	1	36	60.3	66.4	5.4	26.5	12.9	61.9

APPENDIX H

ENERGY END USE DATA

BETTY BRITE DRY CLEANING AND LAUNDERY PLANT
ELECTRICAL ENERGY CONSUMPTION BY END USE (kWH)

DATE	OPERATING DAY	SPINNER	WASHING MACHINE	AIR COMPRESSOR	HOT WATER TANKS	DRY CLEANER	DRYER	TOTALS
JAN 19	THURS	0	0.2	1	48.6	0.5	1.5	51.8
JAN 20	FRI	0	0	0.8	44.8	1.5	1.6	48.7
JAN 21	SAT	0	0.2	2.1	74.6	0.9	4	81.8
JAN 23	MON	0	0.1	1.4	56.4	0.9	3	61.8
JAN 24	TUES	0	0.2	0.1	99.1	0.1	0.1	99.6
JAN 25	WED	0	0.2	1.7	51.7	0.7	1.1	55.4
JAN 26	THURS	0	0.1	1	49.7	1.4	4.2	56.4
JAN 27	FRI	0	0.2	1.2	54.3	0.3	0.8	56.8
JAN 28	SAT	0	0.1	0.5	45	0.9	1.7	48.2
JAN 30	MON	0	0.1	0.3	47.6	0.7	2	50.7
JAN 31	TUES	0	0.3	1.8	85.1	1.7	3.9	92.8
FEB 1	WED	0	0.1	0.5	40.4	1.4	3.1	45.5
FEB 2	THURS	0	0.2	1.7	35.4	0.8	0.1	38.2
FEB 3	FRI	0	0.1	0.3	49.9	1.2	3.3	54.8
FEB 4	SAT	0	0.1	0.4	36.6	0.2	1.4	38.7
FEB 6	MON	0	0.3	2.2	56.4	0.3	2.2	61.4
FEB 7	TUES	0	0.2	1.2	61.2	1	1.8	65.4
FEB 8	WED	0	0.2	0.1	60.1	0.7	1.5	62.6
FEB 9	THURS	0	0.1	0.8	59	0.9	2.2	63
FEB 10	FRI	0	0.1	1.9	58.4	0.8	1.7	62.9
FEB 11	SAT	0	0.1	0.8	42.6	0.7	2.7	46.9
FEB 13	MON	0	0.3	0.7	75.7	1.1	2.7	80.5
FEB 14	TUE	0	0.2	1.1	50	0.8	1.9	54
FEB 15	WED	0	0.2	0.8	48.5	0.6	0.9	51
FEB 16	THU	0	0.1	1.2	51	1	1.8	55.1
FEB 17	FRI	0	0.2	1.3	62.3	0.9	6	70.7
FEB 18	SAT	0	0.2	1.3	50.1	1.1	1	53.7
FEB 20	MON	0.8	0.2	0.9	69.9	0.8	1.1	73.7
FEB 21	TUE	0.8	0.2	0.9	53.7	0.9	2	58.5
FEB 22	WED	0.5	0.3	2	56	0.8	1.9	61.5
FEB 23	THURS	0.7	0.1	0.5	48	1.6	1	51.9
FEB 24	FRI	0.6	0.2	0.9	121.6	0.7	1.2	125.2
FEB 25	SAT	0.6	0.2	1	105.9	0.6	1.9	110.2
FEB 27	MON	0.5	0.2	1.4	49.7	0.7	2	54.5
FEB 28	TUES	0.6	0.1	0.9	49.7	0.5	2	53.8
FEB 29	WED	0.6	0.2	1.2	49.7	0.6	2.3	54.6
MAR 1	THURS	0.5	0.1	1.3	96	1.2	2.3	101.4
MAR 2	FRI	0.1	0.1	0.4	32.4	0.4	1.3	34.7
MAR 3	SAT	0.5	0.1	0.8	94.2	0.8	2.1	98.5
MAR 5	MON	0.8	0.2	1.3	88.1	0.5	3.5	94.4
MAR 6	TUES	0.5	0.2	0.9	117.1	0.6	1.1	120.4
MAR 7	WED	0.7	0.2	0.9	119	1	2.7	124.5
MAR 8	THURS	0.5	0.1	1.2	55.1	0.7	1.5	59.1
MAR 9	FRI	0.4	0.3	1.8	89	1.1	2.3	94.9
MAR 10	SAT	0.5	0.1	1.1	39.1	2	2	44.8
MAR 12	MON	0.7	0.2	0.2	29.3	1	3.5	34.9
TOTAL		10.9	7.7	47.8	2858	39.6	95.9	3059.9
DAILY AVG.		0.57	0.17	1.04	62.13	0.86	2.08	66.52